POWER

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Expert

Some wise person has said, "An expert is an ordinary man away from home."

Humorous, yes—but there is more than a grain of truth in this definition. It emphasizes one of the most valuable attributes of the expert who is called upon to give judgment in engineering problems.

Because the consultant is "away from home" and free of the traditions and inhibitions which ofttimes hamper the thinking of the men on the job, he is frequently able to see more clearly and offer valuable advice.

For instance, a consulting engineer was invited to pass upon the merits of three alternative plans for the solution of a vexatious power problem. He found a fourth solution that was more economical than any of the others.

And great was the joy and emolument thereof!

Now, this was no reflection upon the makers of the rejected plans. No one thought it so. It simply demonstrated the practical dollars-and-cents value of the "man away from home."

EDITORIALS

Servicing the Chicago Post Office

It has always been one of *Power's* cardinal editorial policies that power shall be made when it should be made, and bought when it should be bought. Between these two extremes there is plenty of room for engineering discussion and difference of opinion. Undoubtedly, specific engineering conditions must apply to individual cases, and the economics upon which either conclusion is based must, in the last analysis, rest with the person who holds the deciding vote.

As an aid to better understanding of the problems involved in decisions of the sort, the article discussing conditions in the new post office under construction in Chicago is a valuable contribution. The author, a well-known consulting engineer of Chicago, has gathered together such information as he could in connection with the contracts recently awarded for the power and heating services for the new post office, and has balanced their costs with similar services generated and delivered by a plant installed for the express purpose of supplying this one building.

Power hopes that the article will be carefully read, and that arguments and discussion from both sides may be developed, as they will be the means of a better understanding and clarification of the two divergent viewpoints involved, and a constructive aid in the making of future decisions in which the same thoughts require consideration.

Better Breakdown Rates Required

To some industrial companies the question of what shall be done to assure continuous power service, should their own fail, is of vital consequence.

There are several answers to the question and each should be given careful thought. In many cases the provision of supplemental auxiliary equipment reduces the power shutdown hazard to a negligible degree. In others the solution may be simplified by arranging with some near-by utility to furnish emergency service in the event of necessity.

Unfortunately, many industrials which would avail themselves of this class of service are estopped by the exorbitant or, at least to them, impossible rates. It almost seems as though some of the utilities were bent upon making their stand-by emergency service rates proof positive that the industrial had no business attempting to make its own power—since to do so would inevitably subject it to a heavy penalty for emergency service.

"It should, of course, be obvious that purchased power," the utilities might say, "really effects a great saving, because it eliminates the need for connecting an emergency service."

Aside from the fact that such an argument is full of holes, the policy of holding up the price of emergency service really works against the best interests of the utility.

The actual call for emergency service is so seldom, and even then of so little consequence to the load capacity of the utility, that its initial cost is practically confined to the actual expense of labor and material for bringing the power leads to the industry's property.

If some plan were worked out whereby the industry would pay a reasonable rate of return on the utility's connection cost, as a flat monthly rate and an addition for the power used at some normal rate, two things would be accomplished. First, the utilities would receive a proper return on their capital investment as well as for any power used. Second, because the industrials would then have an economical stand-by arrangement, they would be more inclined to use it.

Waste Steam for Air Preheating

Many power plants that during the winter months utilize all of the steam exhaust because of heating demands, lose a considerable amount of steam to the atmosphere during the summer. Some useful purpose should be found for this steam so that its waste can be reduced.

A use which seems to be largely overlooked and which should result in increased boiler efficiency is that of preheating air for combustion. Assuming eighteen pounds of air are required per pound of coal burned, about 560 B.t.u. are required to raise the temperature to 200 deg. This will employ slightly more than a half pound of exhaust steam for each pound of coal consumed. Preheating the air to this temperature will increase the boiler efficiency and thereby decrease the fuel used.

Only a few central stations have found it expedient to use bled steam for air preheating, because the availability of the steam for power generation adds to the cost of air preheating. But where normally wasted exhaust is used the only cost is the fixed charges on the air-heating equipment. In this case it would seem that steam air preheating should yield attractive returns and be worth investigating.

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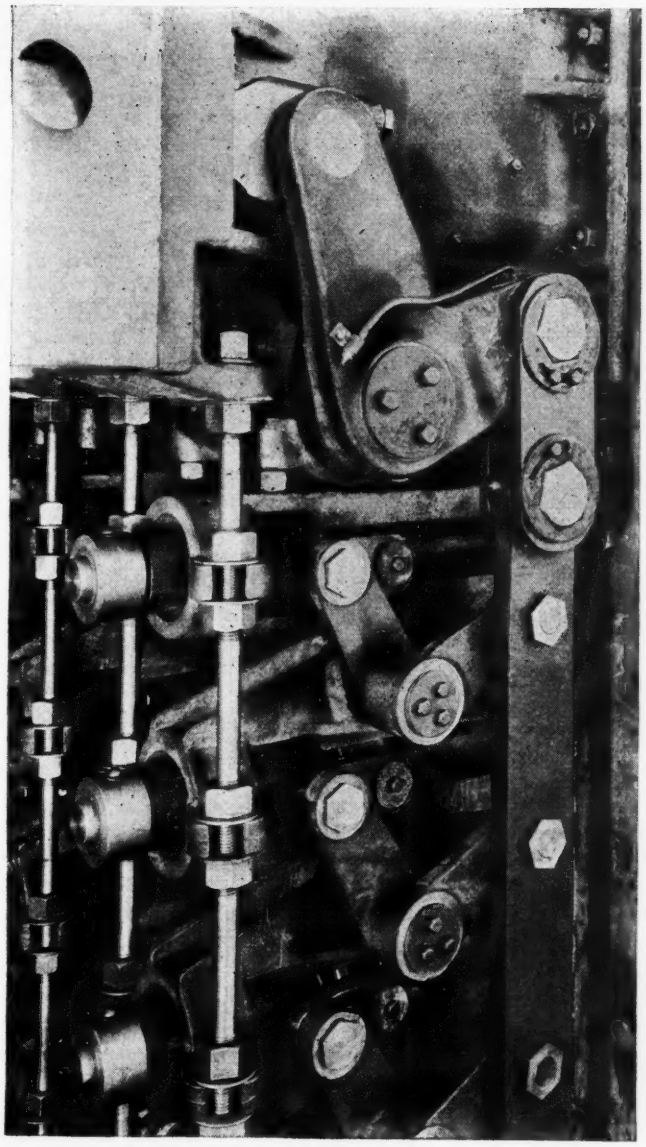
POWER Stands for . .

1. Making Power When It Should Be Made
2. Buying Power When It Should Be Bought
3. Cheaper Power Through Modern Equipment
4. Easier Financing of Equipment Purchases
5. Better Use of Byproduct Heat and Power
6. Operating Methods That Save Money
7. Less Waste in Transmission and Application
8. Prevention of Smoke, Within Reason



Pusher rods allow individual stroke adjustment

Zoned Air Control For Stokers Made Automatic

By F. J. CHATEL
Detroit Edison Company

Individualized windboxes with individual control of air flow for the distribution of the air to each retort of underfeed stokers have been in operation for some time at the Beacon Street heating plant of the Detroit Edison Company. Regulation of air flow to the various windboxes was manual, as described in the June 23, 1931, number of POWER. A similar system, improved in the light of previous experience, and with air flow control made completely automatic has been installed on the stokers at Delray No. 3. Improvements in operating efficiency are more noticeable in normal operation than during tests

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INVARIABLY the designers of new power plants, in their aim to better past performances, incorporate ideas necessitating new mechanical designs. These designs often present developments which must necessarily be refined after the equipment is in operation and some indication of its performance is obtained. It was expected that Delray Power House No. 3, with its unusually long single-ended stokers operating with 350 deg. F. preheated air would involve several new developments which would need further attention. The stoker, fifteen retorts wide, has fifty-seven tuyères per retort and a total projected grate area of 611 sq.ft. In the original design there were six pushers to the retort, each having individual stroke adjustments with a maximum stroke of eight inches. The extension grates are divided into an upper movable and a lower stationary section, both supplying preheated air to complete the combustion.

One of the problems which confronted the designers was the proper distribution of air to a stoker of this size. It was realized that it would be impossible to observe the fuel bed with the accuracy required for regulating the air flow to all sections. With coal having a high volatile content, it was evident that there would often be flames which would obscure parts of the fuel bed. It was also desired to minimize the number of access and observation doors so that there would be the least amount of interference with the water walls. With these points in mind, it was concluded that some form of metered air control should be installed in preference to the conventional manually operated dampers which give no quantitative indication of changes in air volume.

Considerable thought had been given to metered air control either automatically or manually operated, but there were no installations in operation up to this time. Preliminary experimental automatic installations were made at the Marysville plant, and although these were not entirely successful, they led to the design of the automatic air control installed at Delray No. 3. This control has now passed the development stage and although refinements are still being made it has become a desirable installation in connection with large stokers. As a comparative development, manually operated metered air control was installed at the Beacon Street

heating plant. This control was described in the June 23, 1931, number of *Power*.

Automatic Air Control

The tuyère section of the Delray No. 3 stoker was divided into four general zones, and an individual windbox and air duct supplies preheated air to each of the four sections on each retort, making 60 such windboxes for one stoker. This division is shown in Fig. 1.

Referring to Fig. 2, the windbox consists of a sheet-metal box in which are placed an orifice and a control damper. All these dampers are similarly constructed and are of the balanced cylindrical type. Air enters at each end and is throttled through the slots in the cylindrical surfaces. The orifice does the metering and the control damper regulates the air volume.

The pressure differential across the orifice, a measurment of the air flow volume, is maintained practically constant throughout the operating range of the boiler. This is accomplished by varying the area of the orifice in proportion to the load demand. All orifices are linked together and operated simultaneously by a mercury-sealed motor mechanism, *A*, Fig. 2, which has for its motive force the differential between the windbox and atmospheric pressures. Counterbalance weights W_1–W_2 are adjusted to obtain orifice openings in proportion to the windbox pressure.

Each control damper is operated by an individual motor mechanism which is balanced by deadweight W_3 for a predetermined orifice differential. Any change in air flow due to change of fuel bed thickness causes a corresponding change in orifice differential. This change of differential pressure unbalances the motor mechanism *B*, Fig. 2, causing it to operate the control damper until the original differential pressure is restored, thereby maintaining a constant air volume. The orifice discharge pressure is exerted on top of the motor mechanism bell,

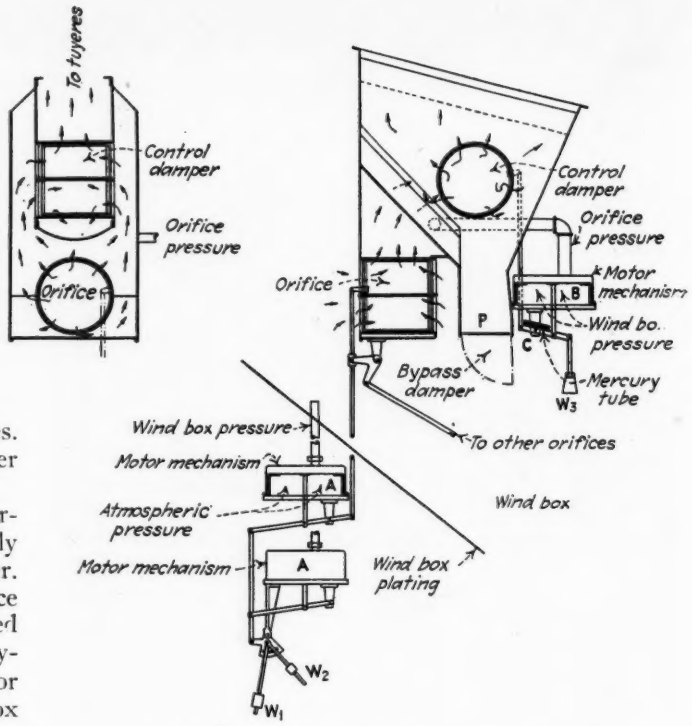

Fig. 2—Separate control mechanism regulates air on each windbox

while the windbox pressure is exerted on the opposite side. The amount of air flow through the control dampers can be varied by changing the weight W_3. With the orifice in the wide-open position, the differential, with full windbox pressure corresponding to maximum rating of the boiler, is $1\frac{1}{2}$ in. of water. This determines the operating orifice differential and the amount of weight W_3 required to maintain the control damper balanced. This orifice pressure drop represents additional windbox pressure with a corresponding increase in fan load, which is required with the automatic air control.

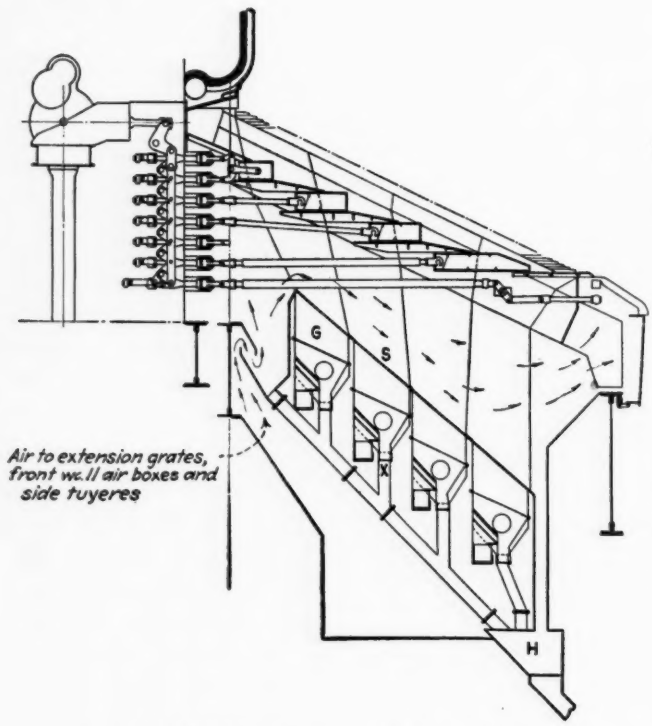

Fig. 1—Individual windboxes divide each retort into four zones

Operation

With a fuel bed of the desired thickness over the control damper the latter will remain balanced. Should the fuel bed become heavier or thinner, air flow will increase or decrease and the orifice pressure drop will increase or decrease, resulting in a pressure differential greater or less than $1\frac{1}{2}$ in. of water exerted in bell *B*. The difference between this resultant pressure and $1\frac{1}{2}$ in. of water momentarily unbalances the motor mechanism which checks or opens the control damper until the balance is again established. However, if the control damper remained balanced under these conditions it would supply a constant air volume and the fuel bed would not regain its original desired thickness. To overcome this, a tube *C* partly filled with mercury is placed directly over the fulcrum of the damper-operating lever. When the unbalancing occurs and the lever tilts, the mercury flows to the low end, thereby increasing the motive force causing the damper to overtravel, which decreases or increases the air volume sufficiently to allow the fuel bed to regain its proper thickness.

The operating principle of this automatic air control is quite similar to the manually operated metered air control installed at the Beacon Street heating plant. With the latter, both the orifice and control dampers are operated manually. This feature has an advantage in that it

requires less orifice drop and therefore a lower windbox pressure.

The automatic control responds to fuel bed changes instantaneously, while the manual control must rely upon thorough fuel bed observations, or a manometer for each orifice to obtain an indication of the flow of air through each control damper. In either case, the operator must be constantly watching for fuel bed changes and making corresponding damper adjustments.

Although the automatic air control needs no air pressure gages to determine its damper settings, it has been found desirable to install a limited number of underfire pressure gages as an aid to correcting poor coal distribution. With individual pusher adjustments capable of changing the pusher strokes as much as eight inches, it is possible to feed coal to any stoker section at a rate greater than the maximum air control range of the dampers. A control damper going from closed to open position will increase the air flow about 10 per cent, but will always remain open if the pusher adjustment is such that the increased amount of coal fed over this damper requires a corresponding increase of air to burn it. If the pusher is poorly adjusted enough coal may be fed to require more than a 10 per cent air increase in which case the fuel bed will continue getting heavier with the control damper in the open position. To detect this condition twenty underfire pressure gages are installed to enable the operator to correct poorly adjusted pushers so that they will at all times be within the range of the control dampers. One pressure gage in each zone of damper control boxes for each section of the stoker has been found sufficient for this purpose. These gages will also indicate excessive clinker formation which will produce the same effect.

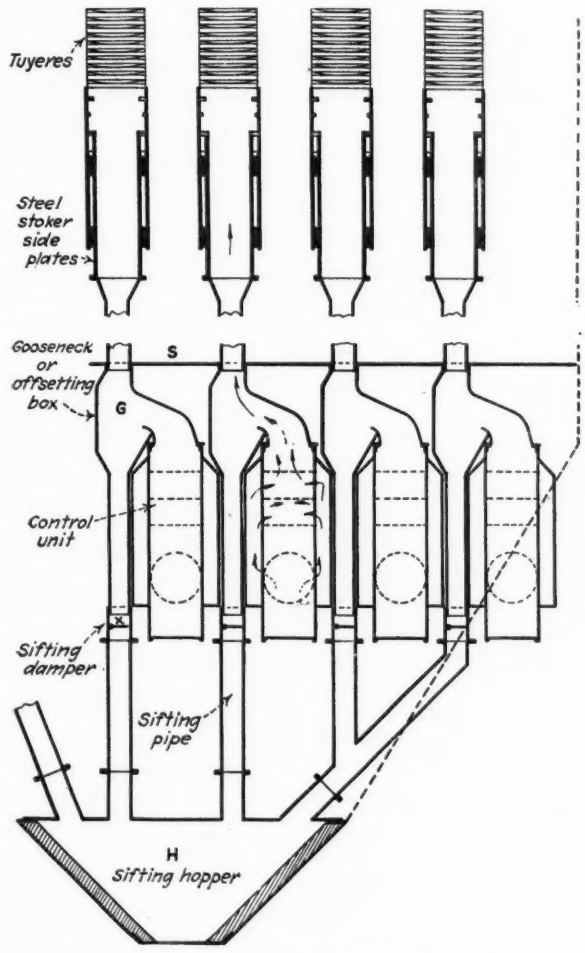

Fig. 3—Offset goosenecks keep siftings out of windboxes and dampers

The control was originally designed to supply air through six zones or rows of windboxes which were to be suspended from the steel stoker sideplates directly beneath the tuyères. The first installation of this kind proved conclusively that this was not practical, as coal siftings quickly fouled the dampers. It was then decided to install them in connection with a system of ducts which would isolate the moving parts from coal siftings, Figs. 1 and 3. These ducts, attached to the stoker support plates, carry gooseneck or off-setting boxes, to which the windboxes are bolted. Coal siftings from the pushers drop on to shedding plates *S* and then into the hopper, *H*. Siftings from the tuyères drop against the air flow through the main air ducts and goosenecks and are discharged into the same hopper through 4x6-in. pipes. The shape of the gooseneck air outlet is such that high air velocities at this point prevent the falling of coal siftings into the control or orifice dampers. Sifting valves *X* in the pipe ducts prevent air flow and pressure equalization between units. These are operated from the outside of the plenum chamber. Dampers *P*, Fig. 2, at the bottom of the boxes, also operated from the outside, serve as air control bypass dampers. To observe the orifice and control dampers when the boiler is steaming, the plenum chamber is illuminated and port holes are placed in the front casing.

Extension-Grate Air Not Controlled

Installing these control units in connection with this duct system eliminated coal sifting difficulties, which made the installation practicable but required more space. It was decided to reduce the number of zones from six to four rather than make extensive plenum chamber changes. This change, however, increased the pressure drop through the orifice, as the air required for maximum flow has but two-thirds of the original air flow passages. In future installations this pressure drop, or resistance, can be reduced by altering the design of the orifice and control dampers so that the air flow passages will be of the proper size.

No control units have been installed for metered air supply to the extension grates because of design changes being made to the latter. The extension grates were the weakest part of the stoker with relation to iron burned and it was believed that slight changes combined with more experience in handling the clinker formation over them would minimize the burning of iron so that it would become practical to install an air control below them. Unfortunately, at the present time the beneficial effects of the air control under the tuyères are often lost by improper air distribution through the extension grates.

Under test conditions, with and without the air control, a slight advantage is found with the air control in operation. However, under operating conditions with similar stokers burning the same coal and operated by the same operator through a long period the air control usually shows a decided improvement. The small difference obtained with and without the air control during test periods is a result of the operator's maintaining definitely fixed pusher adjustments for any given rating and observing conditions over the extension grates more closely than can be expected during daily normal operation.

It is always difficult to show the true benefits from a mechanism of this kind by tests, because the skill of the

operator is really the determining factor. Under test conditions, a skilled operator by close attention can do almost as well without the air control. The real benefit will be proved by monthly results when all boilers are equipped with the controls. Thus far it appears that a material increase in efficiency will be obtained.

Stoker Developments

As experience was obtained with the stoker, it became evident that other changes could be made to improve combustion and reduce maintenance costs. The stoker originally had six sets of square-nosed pushers operated in conjunction with each ram to push the burning coal, coke and ashes toward movable extension grates. Although the bell crank mechanism through which the pushers operate was designed for individual pusher and extension grate adjustments, made externally, it was found difficult to maintain a properly shaped fuel bed at the ram box and in the retorts. Coal at the stoker front and in the retorts was not sufficiently packed, and excessive agitation resulted in smoke. Combustion often took place too deep in the retorts, resulting in the burning of iron and the formation of large clinkers over the bottom pushers and extension grates.

To improve these conditions lower ram box caps were installed and the bottom pushers replaced with partly beveled pushers six inches longer. This improved the fuel bed at the top and bottom zones and is an arrangement which can be used quite successfully with some grades of coal. It was evident, however, that the number of pushers would have to be reduced to obtain the desired results with all grades of coal. Fewer pushers would permit shorter strokes or greater stroke reductions from top to bottom of the retorts, resulting in more packing and less agitation. Inasmuch as the proper number to be used was questionable, steel side cover plates were installed to obtain straight retort sides so that the number and location of pushers could be changed without other retort changes. It was decided to convert one stoker to a five-pusher and another to four, the two to be compared. In either installation, the bottom pusher was to be especially long and partly beveled to remove clinkers effectively without handling an excessive amount of coke.

Although the five-pusher arrangement showed slightly improved conditions, indicating that it was a step in the right direction, the four-pusher showed a greater improvement, and other stokers are being converted to this arrangement. The deepened retort increases coal packing, resulting in a more uniform and controllable fuel bed. It also reduces agitation and prevents burning of coal too low in the retorts, which invariably is the cause of clinker formation and the burning of iron. Four rows of pushers also conform with the four zones of air control, resulting in less frequent operation of the dampers. The efficiency of the unit has been increased and iron replacements reduced. Coal siftings from the pushers are less than one-tenth of 1 per cent, and decreased friction has slightly reduced the power required to drive the stoker.

The extension grates, with the parts immediately above and below them, were more susceptible to burning than any other part of the stoker. These grates, which consist of bars with fins which form ports through which the air flows, were originally held in place by hooks at the top and bottom. Raising the lower hook seven inches from the bottom and redesigning the top grate guard castings to permit free air flow to the lower section of the grates has reduced the burning appreciably. Experiments with other types of extension grates that reduced the air openings failed to show improvements in economy or handling of the ashes. Although these grates do not require as much air as the grates of double-ended stokers, it is evident that a considerable amount must pass through them to complete combustion.

With the improved pusher and extension grate installations it is not unusual to operate the stoker from two to three months before it is shut down for inspection or repairs. Under these conditions the hand-operated dampers now in use with the extension grates may safely be replaced with metered air control dampers, which will undoubtedly further improve conditions, particularly at low ratings.

Monthly over-all boiler efficiency shows a trend in relation with the pusher changes and zoned air control installations. There are five boilers installed, three of which have been in operation normally since the plant went into regular service late in 1929. Although experience in handling the original as well as the new arrangement has played some part in increasing the over-all boiler efficiency, a marked improvement in results was obtained as improved stokers went into operation. Extreme fluctuations in the over-all boiler efficiency from month to month are generally due to errors in the bunker readings, which are automatically corrected with the succeeding monthly readings. Delray Power House No. 3 assumed its share of the system load from the start and has continually operated with an average boiler rating of 240 per cent (based on total boiler surface, including water walls) and a ratio of banked to steaming hours of 22 per cent.

Over-all boiler efficiency and continuity of service being obtained with the improved stokers and zoned air controls indicate that the final unit meets the requirements. It is planned to incorporate the changes described on all the stokers.

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Thermionic Tubes in Industry

IN THE application of thermionic tubes to control purposes, the year 1931 recorded gratifying progress, according to *Electronics*. Steel mills, sheet plants, wire mills, rubber factories, paper machines, chemical processes, and many other industrial operations are making wider use of tubes.

Light control by thermionic devices on an increasing scale was applied in theaters, auditoriums, ball-rooms, building flood lighting, etc.

Elevators are leveled by three-element tubes, railroad signals are tube-operated, and train-stop apparatus is actuated by thermionic apparatus that shows distant signal settings directly in the locomotive cab.

Increasing attention is being given to handling heavy currents by tubes. Direct-current transmission over underground cables in cities is being seriously figured by lighting companies. A tube installation will shortly supply part of Boston's through-wire Edison direct-current section directly from the alternating-current system.

Figures presented before a New York meeting of the American Institute of Electrical Engineers in December testified that the economic distance for water-power transmission can be multiplied nine times by using direct-current as against present alternating-current methods.

Architect's drawing of the new Chicago Post Office

Why Not a Power Plant For the Chicago Post Office? . . . *ASKS*

C. M. GARLAND
Consulting Engineer
Chicago

The question of whether power and steam should be purchased or generated on the site always engages attention. When the contracts for these services in the Chicago Post Office were let by the government, a study of the economic possibilities of government operation of its own plant was made by the author, who concluded that approximately $145,850 yearly might have been saved

THE Treasury Department at Washington recently issued specifications covering the construction of a new post office building at Chicago which has been heralded as an example of unusual efficiency. The contract on this building has recently been let and work is now in progress. The building contains no space for a boiler plant or a power plant, and the specifications state that the electric energy will be purchased from the Commonwealth Edison Company and that steam will be purchased from a heating company. The heating company selected for this service is the Union Station Company, of Chicago.

On learning the size of the building, and that the heat and power was to be purchased, the writer's curiosity was aroused. Those conversant with the subject know that where any large low-pressure steam load exists the power can be generated at a low cost. As an engineer-taxpayer, therefore, the author has investigated the matter of heat and power for the new post office building, and the report of this investigation follows.

The new building will be twelve main stories in height, 796 ft. long and 344 ft. wide. The total cubic contents of the building is something over 50,000,000 cu.ft., and the total floor area approximately 3,500,000 square feet.

The specifications on heating and ventilating do not state the basis of the design of this system, nor do they give any specific information on the total power and heating requirements. From the size of the pumping equipment specified, however, it would seem as though the system were designed to take care of the equivalent of about 600,000 sq.ft. of direct radiation, or approximately 150,000 lb. of steam per hour. From the cubical contents of the building, a rough estimate shows that the steam requirements per hour at —10 deg. outside air, and with a room temperature of 70 deg. at the breathing line, would be approximately 120,000 lb. of steam per hour. However, to take care of all the different heating requirements it has been assumed in the figures which follow that the maximum steam consumption with—10-deg. weather will be 150,000 lb. per hour, and from this assumption a total steam consumption during the year of 376,500,000 lb. of steam has been calculated from the mean monthly temperature in Chicago. An average steam consumption per hour for the month of May is estimated at 22,500 lb. and for the month of January of 86,400 lb. For the summer months the steam consumption has been estimated at approximately 15,000 lb. of steam per hour for different heating operations around the building.

In the Chicago territory during a normal year it is necessary to supply steam for heating purposes from Sept. 15 until May 15, and not infrequently it is necessary to supply this steam until June 1. Calculations indicate that there should be enough heating steam to generate power from Oct. 1 until May 1. The estimate of 15,000 lb. of steam per hour for heating operations

during the remaining so-called summer months is purely an assumption. However, if no steam were required for heating during the summer, approximately 2,640 tons of coal would be needed for generating the steam for power. This, of course, is an extreme condition that will not obtain. In the figures for the cost of coal to cover these summer months and the generation of power it has been assumed that 1,320 additional tons of coal would be required. Considering the size of the building and the miscellaneous use of steam for pumps, for washrooms, steaming mail bags, restaurants, etc., it is believed that this is a conservative estimate. Even though there were no steam requirements for heating purposes during the entire summer months the additional coal cost would be $3,300, which is an item of comparatively small importance.

Boiler Plant

From the above figures a tentative boiler room design suitable for supplying heat for this building has been made, on the assumption that the steam would be sent through non-condensing for the generation of power before being discharged to the heating system.

This boiler room includes three steam-generating units, each consisting of boiler, stoker, superheater and air preheater, designed to burn Illinois screenings. Each unit has a capacity of 75,000 lb. of steam per hour, so that two units would handle the peak-load conditions of 150,000 lb. of steam per hour when operated at about 320 per cent rating. In this layout the boilers were designed for 450 lb. pressure, preheaters for 400 deg. F., superheaters to deliver steam at a temperature of 825 deg. F. and air-cooled furnace walls, as smokeless combustion is essential. From the above the total cost of the boiler plant was estimated as follows:

Three 7,500-sq.ft. boilers, etc. erected	$250,000
Coal-handling equipment	30,000
Pumps and heaters	7,500
Piping and covering	50,000
Foundations and excavations	6,000
Water treatment	7,000
Building, 250,000 cu.ft. at 30c.	75,000
Instruments	3,000
Total	$428,500
Engineering, at 6 per cent	25,710
Total cost of boiler room	$454,210

The operating cost of this is estimated as follows:

Three firemen, at $1,800	$5,400
Three helpers, at $1,500	4,500
Three coal handlers, at $1,500	4,500
Coal, 23,000 tons, at $2.50 per ton	57,500
Stokers repairs, at 4c. per ton of coal	720
Repairs to brickwork, at 10c. per ton of coal	2,300
Repairs to boiler pumps, etc. at 6c. per ton of coal	1,380
Supplies, oil, waste, etc.	2,500
Water	1,000
Ash removal	1,500
Total annual operating cost	$81,300

The total tons of coal used is based on a year-in-and-year-out boiler efficiency of 80 per cent. Considering that preheaters have been included, the author believes this to be a conservative figure. It is contemplated that Illinois screenings containing 11,500 B.t.u. per pound costing $2.50 a ton will be used.

According to the Treasury Department the rate for steam is 51c. per 1,000 lb., based on coal delivered to the Union Station at $2.50 per ton, with an increase or decrease of 1c. per 1,000 lb. for each 15c. per ton on the cost above or below the base price.

With the above information and the estimate of 376,500,000 lb. of steam a year, the annual heating bill for an average year will be $192,000. The operating cost for the generation of this steam is shown to be $81,300, so that the loss in operation when purchasing steam is $110,700 a year. Fixed charges will be considered later.

The Post Office requires power for lights, elevators, fans and exhausters for the heating system, and for the operation of conveyors carrying mail. The specifications make no summary of the power requirements, so an estimate has been made as follows:

	Demand, Kw. 15 Min. Maximum
Lights	1,460
Elevators and conveyors	540
Fan motors	1,000
Total	3,000

The total kilowatts required for lights have been based on a lighted floor area of 3,285,000 sq.ft., and a constant of 0.6 watt per square foot and an assumption that 75 per cent of them will be in use at one time. One of the electrical contractors estimated the maximum demand at 4,000 kw., but in my opinion this figure is high. The energy consumption of the building during winter months is estimated as follows:

	Kw.-Hr. per Month
Conveyors and elevators	220,000
Lights	288,000
Fan motors, etc.	288,000
Total kilowatts per month, 8 months in the year	796,000

For the summer months it is believed the average monthly consumption will be 500,000 kw.-hr. On the above basis, and from the Commonwealth Edison schedule, the total annual power bill would be $95,768, and the average cost of energy per kilowatt-hour 1.132c. This figure checks well with other figures examined in Chicago. The average load 24 hr. a day during the winter months will be approximately 1,100 kw., and during the summer months approximately 700 kilowatts.

A tentative design for a turbine plant operating at 425 lb. pressure at the throttle and a total steam temperature of 825 deg. F. has been made. Its installation is assumed to consist of three non-condensing turbine units, two of 3,000 kw. capacity, and one of 1,000 kw. capacity. The table below is an estimated cost of this turbine installation:

Two 3,000-kw. units	$75,000
One 1,000-kw. unit	25,000
Instruments	2,500
Switchboard and cables, erected	18,000
Foundations for equipment	4,000
Cooling tower, cooling water for glands, etc.	3,000
Piping and covering	15,000
Building, 135,000 cu.ft. at 30c.	40,500
Erection of equipment	6,000
Total	$189,000
Engineering, at 6 per cent	11,340
Grand total	$200,340

The operating cost of the turbine room is estimated as follows:

Three attendants, at $1,800 a year	$5,400
Supplies, maintenance and repairs	2,500
Water	1,500
Assistant chief engineer	2,400
Additional coal for power, 1,320 tons at $2.50	3,300
Total	$15,100

In the above figures the salary of a chief engineer has not been included, as he will be required whether power and heat is installed or not. From the estimate on the steam consumption for the eight months in winter the demands vary from 22,500 lb. per hour to 86,400 lb. per hour, while the average turbine load will only be 1,100 kw. The average water rate of the turbine plant will be about 20 lb. per kilowatt hour, which includes steam for pumps and auxiliaries, together with losses in the pipe lines, the actual water rate of the turbines being

approximately 17.3 lb. of steam per kilowatt-hour at full load. The average load on the power plant during the summer months was estimated at 700 kw., requiring 14,000 lb. of steam per hour. About 15,000 lb. of steam per hour will be required for different heating operations around the building during the summer. However, to take care of any discrepancy between the steam required for heating and for power as previously discussed, there has been included in the estimate on the cost of generating power 1,320 tons of coal at $2.50 per ton, or $3,300, for additional coal for power. From the above figures it will be noted that the cost of purchased current is $95,768 a year, whereas the operating cost of the turbine room is $15,100 a year. There is therefore a saving in operations of $80,668.

It is understood that the Treasury Department has estimated the cost of purchased current at about $12,000 a month, whereas our estimate is approximately $8,000 a month. If the Treasury Department's figure be used the annual power bill for the purchase of current would be about $140,000 a year and the operating cost of generated current would be approximately $24,500 a year. The saving in operating cost would then be $115,500 annually.

Fixed Charges

The first cost of the government power plant is as follows:

Boiler room	$454,210
Turbine room	200,340
Total	$654,550

The government obtains its money for about 3½ per cent. The life of a plant of this type under the conditions of service in which the power load very nearly balances the steam load is at least twenty years. There are some plants in Chicago that have been in operation close to forty years under similar conditions. The depreciation therefore, may be taken at 5 per cent, and the depreciation, plus the average interest, allowing for depreciated investment, is approximately 6¾ per cent, or 7 per cent for convenience. The total annual fixed charge, therefore, is $45,518.

It is understood that the government uses a capital charge of 12 per cent on the investment in estimating its costs on power equipment. Considering that the government has no taxes to pay, pays no insurance premiums, and obtains its money for about 3½ per cent, there seems to be no justification in such a charge. The power plant in the present post office building in Chicago has been in service for more than twenty years. These high capital charges are a delusion and show a fictitiously high cost for generated current.

The saving by the generation of steam is	$110,700
The saving by the generation of current is	80,668
The total annual savings in operation	$191,368
The net saving	145,850
(or if the Treasury Department's figures for power are used)	$180,682

The saving in operations will pay for the cost of the plant in less than three and one-half years. Yet the Treasury Department wrote under date of Aug 7, that it would not be economical for the government to install this plant. They further state, "The many years of experience gained by this office in operating buildings under its control, now approximately fifteen hundred in number, has proved an excellent guide in determining whether electricity should be generated or purchased for new buildings being considered, and whether steam should be produced in its own boilers or purchased."

The department has given a number of reasons for not including the power plant. In the first place there was no appropriation. In the second place, considering that the building was built over railroad tracks they could not install a boiler plant. In the third place they claim that ultimately they might install a power plant in the future appraisal building, and then later that the purchase of power and heat is so economical that it would not pay to consider the generation of either.

▼

Comparison Made of the Largest Kaplan Turbines

BECAUSE of the number of large Kaplan turbines installed or under construction in Europe and in the United States, the following table is of particular interest. The table was supplied by Elov Englesson, chief engineer, Verkstaden, Kristinehamn, Sweden. The largest units in dimensions are the two being built for the Vargön plant of the Royal Board of Water Falls, on the Göta River, and are rated at 15,200 hp. under a 14-ft. head. The largest-powered units of this type are the six installed in the United States in the Safe Harbor Water Power Corporation's plant on the Susquehanna River in Pennsylvania. These units are rated at 42,500 hp. under a 55-ft. head. This head is nearly 50 per cent greater than any other installation given in the table.

COMPARISON OF THE LARGEST KAPLAN TURBINES IN THE WORLD

Name of Plant	Where Installed	Turbine Manufacturer	Head Meters	Head, Feet	Turbine Rating, Horsepower	Horsepower Under One-Meter Head	Discharge, Second-Meters	Discharge, Second-Feet	Runner Diameter, Millimeters	Runner Diameter, Feet
Vargon	Sweden	Kristinehamn	4.3	14.1	15,200	1,705	314	11,090	8,000	26.25
Vargon[1]	Sweden	Nohab and Kristinehamn	4.3	14.1	14,800	1,660	305	10,770	8,000	26.25
Swir	Russia	Kristinehamn	11	36.1	37,500	1,025	290	10,240	7,420	24.35
Dogern	Germany, Switzerland.	Escher Wyss	10.7	35.1	35,500	1,015	300	10,600	7,000	22.97
Ryburg[2]	Germany, Switzerland.	Voith, Escher Wyss, Charmilles	11.5	37.7	38,700	993	295	10,420	7,000	22.97
Klingnau	Switzerland	Escher Wyss	7.15	23.46	18,400	963	231	8,160	6,460	21.2
Lilla Edit[2]	Sweden	Kristinehamn	6.5	21.3	11,200	675	146	5,155	5,800	19.03
Safe Harbor[2]	United States	S. Morgan Smith and I. P. Morris	16.8	55	42,500	620	226	8,000	5,590	18.33
Laholm	Sweden	Kristinehamn	8.5	27.9	16,000	645	166	5,860	5,500	18.05
Lanforsen[2]	Sweden	Kristinehamn	9.25	30.4	13,000	461	122	4,310	4,770	15.65
Abborrforen[2]	Finland	Kristinehamn	10.9	35.8	16,000	442	128	4,520	4,560	14.96
Neckargemund[2]	Germany	Escher Wyss	3.97	13.02	3,550	449	80	2,825	4,500	14.76
Milykoski[2]	Finland	Kristinehamn	5.8	19	5,500	426	80	2,825	4,100	13.45
Eching (Isar)	Germany	Escher Wyss	12	39.4	12,000	289	89.4	3,155	4,000	13.12

[1] Guide-vane equipment and governor for this turbine is being built by Nohab. The runner is made by Kristinehamn, who also is responsible for efficiency and output guarantees. [2] In operation.

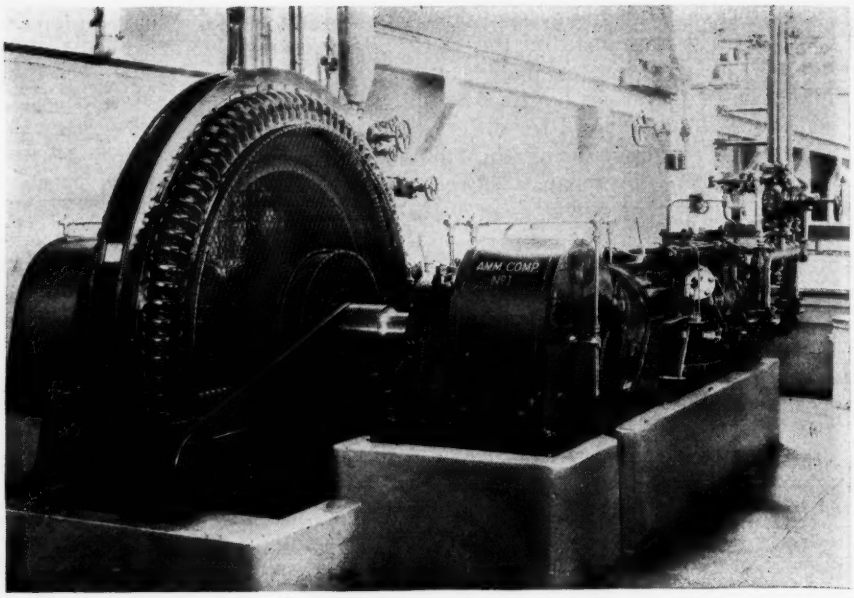

Fig. 1—A 125-hp. 125-r.p.m. synchronous motor driving a two-cylinder ammonia compressor having all the flywheel in the rotor

The Correct Flywheel For Synchronous-Motor-Driven Compressors

By R. C. ALLEN
General Engineer
Westinghouse Elec. & Mfg. Co.

Operation of a flywheel in reducing the current variations to synchronous motors driving compressors and an explanation of the peculiar effects obtained with some flywheels designed by the old methods

IT IS only in recent years that the problem of choosing the correct flywheel for reciprocating compressors driven by synchronous motors has been thoroughly understood. In the past flywheels were chosen which were known to be, or thought to be, ample for the operating conditions. A few exceptions, however, showed that there was something not entirely understood in flywheel applications. A definite knowledge on this subject now makes it possible to select the minimum weight for the flywheel and permits incorporating it compactly as a part of the motor rotor instead of in a separate flywheel.

Flywheel effect of a rotating mass is its inertia value and is expressed as pounds-feet squared, or WR^2, where W is the weight of the flywheel in pounds and R is the radius to the center or gyration. The latter is the radius in feet at which the mass or weight could be concentrated to have the same effect. The desirability of having the wheel diameter large to secure a large R and less weight W is evident.

The function of a flywheel is to help the motor over the load peak or to smooth out a varying load torque. This is done by the flywheel giving up its energy when the load or torque is high or more than the average and absorbing energy when the load is below the average. This means that the load on the motor is smoother where there is a flywheel of ample capacity and the line current taken by the motor is less fluctuating than without a flywheel. There is then less possibility of line disturbance or lights flickering.

The flywheel for a punch press, for instance, is selected to handle the peak load of the punching with about a 15 per cent speed drop. The motor has to deliver that part of the load torque not carried by the flywheel and during the rest period of the operating cycle, return the energy to the flywheel lost by the drop in speed.

A reciprocating compressor is generally driven by a direct-connected synchronous motor. One might think there could be no change in speed with a synchronous motor and no serious variation in the compressor load torque, especially for the two-cylinder horizontal double-acting type. Both of these assumptions are approximately true. A synchronous motor's speed in revolutions per minute is constant, but the speed tends to change with a change in torque. Many compressor torques are anything but constant, as, for instance, for a single-acting machine or for part-capacity operation of a two-cylinder compressor.

The pull of the stator's rotating magnetic field on that of the rotor is much the same as if these fields were connected by a spring that stretched in proportion to the driven load. At full load the spring is stretched to full-load tension. As the load torque changes, due, for instance, to the variation in the compression cycle

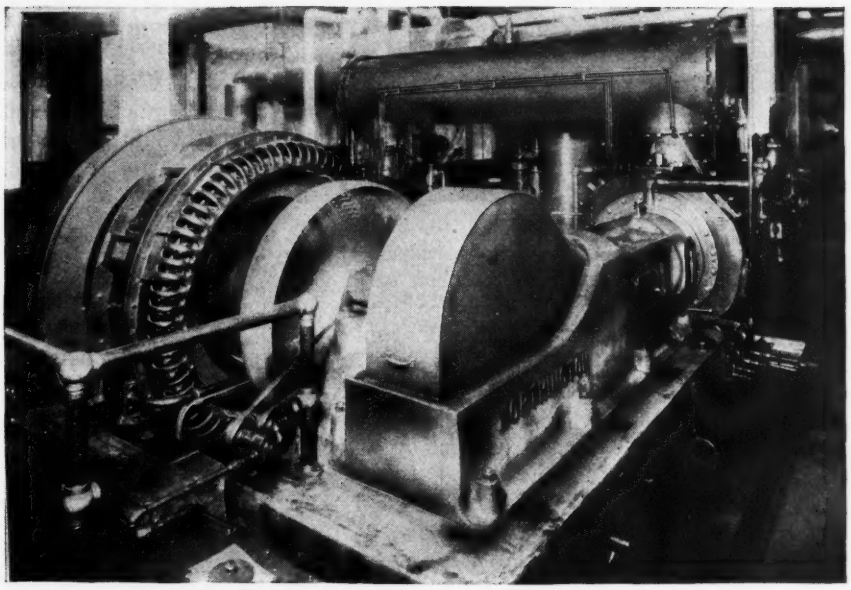

Fig. 2 (Left)—A 500-hp. 187-r.p.m. motor driving a two-cylinder two-stage air compressor having a separate flywheel

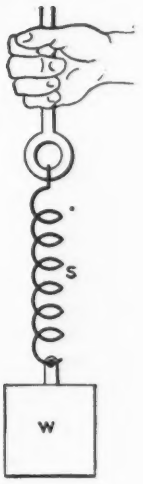

Fig. 3 (Right)—Analogy of a synchronous motor driving a reciprocating compressor

in the cylinder, the spring tension changes in direct proportion. This change in the rotor's angular position with respect to the constant rotating speed of the stator's magnetic poles is a speed change even though the revolutions per minute remains constant. It is to limit this change or the resulting motor current variation that the flywheel is required.

The refrigeration and air-compressor societies, in conjunction with the electrical manufacturers, have established an allowable motor-current variation of 66 per cent, 33 per cent plus and minus. This is based on the motor's full-load current rating, as a standard for compressor drives for any compressor operating condition. A current variation of this value should not cause a line voltage variation sufficient to affect the lighting on a normal power system.

It is the spring effect between the stator and rotor magnetic fields that causes trouble in applying flywheels for motor-driven compressors. Normally it would be expected that any flywheel is better than no flywheel, because it would be of some help; but with a synchronous motor the spring action results, sometimes, in a condition that no flywheel causes less current variation than one sometimes selected by former methods. Of course a still heavier wheel could be used that would reduce this variation to the desired amount.

The reason for this can be understood by referring to the mechanical analogy, Fig. 3. If spring *S* and attached weight *W* are caused to vibrate by moving the hand up and down the movement of the weight may be equal to, less than or more than that of the hand. The movement of the weight would depend upon the relationship of the mass, the frequency of vibration, and the spring constant. With a very slow motion of the hand the weight would follow directly, particularly with a rigid spring. With rapid movement of the hand the weight would remain practically stationary, particularly if the spring were very flexible. For a certain rate of movement of the hand and a value of weight and spring constant the movement of the weight could exceed that of the hand. This is due to the resonance of the system, spring and weight, to that of the vibrating force of the hand. The resonance of a bridge to a trotting horse is a similar example.

In a synchronous-motor-driven compressor the vibrating force corresponding to the hand is the varying torque

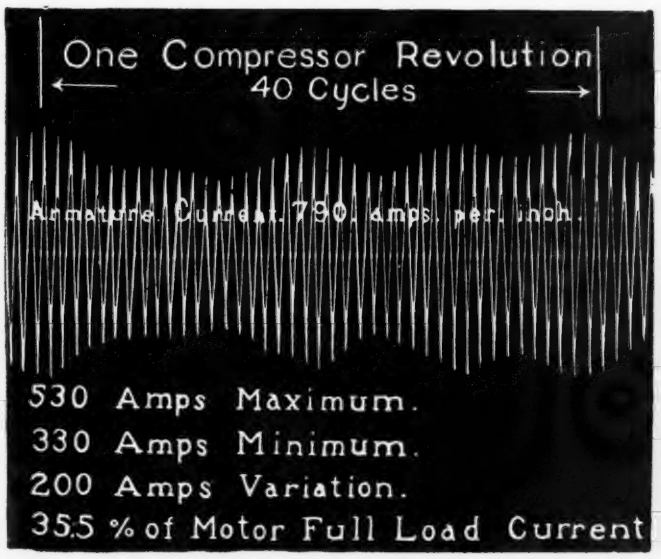

Fig. 4—Oscillogram showing the current variation in a synchronous motor driving a two-cylinder ammonia compressor at full load

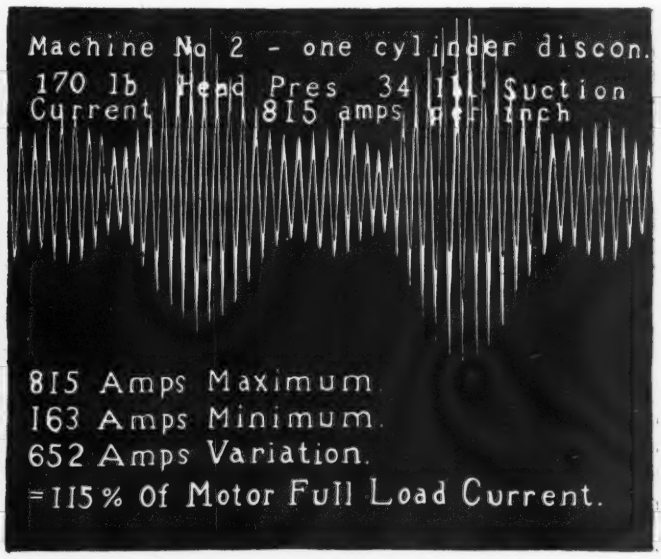
Machine No 2 - one cylinder discon.
815 amps per inch
815 Amps Maximum
163 Amps Minimum.
652 Amps Variation.
=115 % Of Motor Full Load Current.

Fig. 5—Oscillogram of the two-cylinder compressor, Fig. 4, with one crank disconnected to obtain one-half capacity

required by the compressor and transmitted from the stator to the rotor by the magnetic spring action previously mentioned. The motor design constant, expressed in kilowatts input per electrical degree displacement of the rotor, corresponds to spring S. The flywheel effect WR^2 of the unit (flywheel, motor and compressor) corresponds to weight W. The resonance or natural frequency of the motor system, in beats per minute, is expressed as

$$F = \frac{266{,}500}{R\,P\,M}\sqrt{\frac{P_s f}{WR^2}}$$

where $R\,P\,M$ = Revolutions per minute.
f = Frequency of electrical circuit.
P_s = Motor design constant.

The beat of the compressor is called the forced frequency, and may equal the revolutions per minute for a single-cylinder single-acting compressor, two times

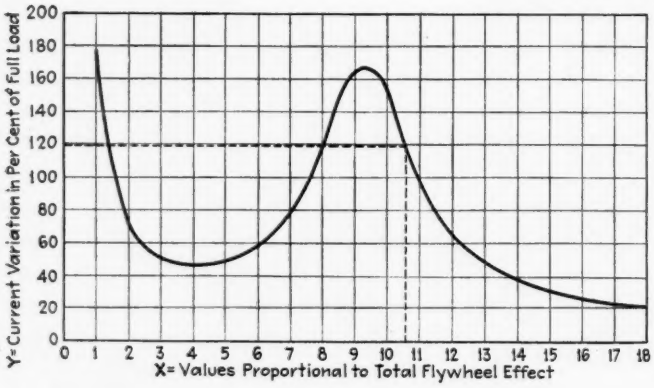

Fig. 6—A flywheel-calculating curve for a two-cylinder ammonia compressor

the revolutions per minute for a two-cylinder single-acting compressor, etc. With a two-stage, two-cylinder compressor, or a two-cylinder single-stage compressor, there are important beats that equal the revolutions per minute and four times the revolutions per minute. When the forced frequency is equal to, or close to, the natural frequency there is a condition of resonance, and a higher rotor swing results than that produced by the normal compressor unbalance. This swing is limited by the damping action of the starting winding and is prevented from becoming infinity, theoretically. The further away from the resonance the less this overtravel or the net angular variation.

A compressor directly driven by a motor has only one factor that can be varied to prevent resonance, and that is the flywheel. It is therefore necessary to select a total flywheel effect to limit the angular variation to the desired amount. Angular variation between the rotor and stator gives proportionately current variation in the motor. The test for correct selection of a flywheel is determined by checking the current variation. This is most accurately done by an oscillograph which photographs each alternation of the alternating current and shows the magnitude of each alternation during a compressor revolution.

The current variation for a two-cylinder horizontal double-acting ammonia compressor with a properly applied flywheel is shown in Fig. 4 on the oscillogram. Fig. 5 shows the increased current variation of the same unit when one compressor crank is disconnected to give half compressor capacity. Under this condition 115 per cent current variation is allowable.

In the days when the behavior of the synchronous motor was not as well understood, a flywheel would occasionally be wrongly chosen that would give high current variation. The operator would try removing the wheel only to find the current dropping to normal. The reason for this is now easily understood. Referring to the flywheel curve, Fig. 6, called an XY curve, if the flywheel chosen gave an X of about 10.5, as was likely from the old method of calculation, the motor current variation Y would have been about 120 per cent. Removing the flywheel would change the X value of the unit to approximately around 5 and result in a motor-current variation of about 50 per cent. A larger flywheel to give an X of 13 would, however, have given the same result.

When a compressor is arranged to operate at part capacity a flywheel effect must be chosen that will give satisfactory operating conditions for all loads. This means, of course, that for some of the operating conditions the WR^2 will be larger than required, resulting in a current variation of less than the specified 66 per cent, while for other conditions, or probably only one loading condition, it will be just the right amount.

It is general practice to include all of the WR^2 required by a refrigerating compressor in the motor, Fig. 1; but with air compressors a separate flywheel is usually furnished, Fig. 2. Air compressors, in general, require more WR^2 than can be included economically with the motor rotor, due to the part capacity method of unloading, which gives a more unbalanced torque than that for refrigerating units.

For a while there were numerous methods of securing the flywheel effect in the rotor of the synchronous motor, such as widening the rotor ring the required amount or building a flywheel onto the rotor and extending it up by the stator, Fig. 7, or using a larger-diameter motor frame, or even building the rotor outside of the stator. The outcome of these was a motor designed especially

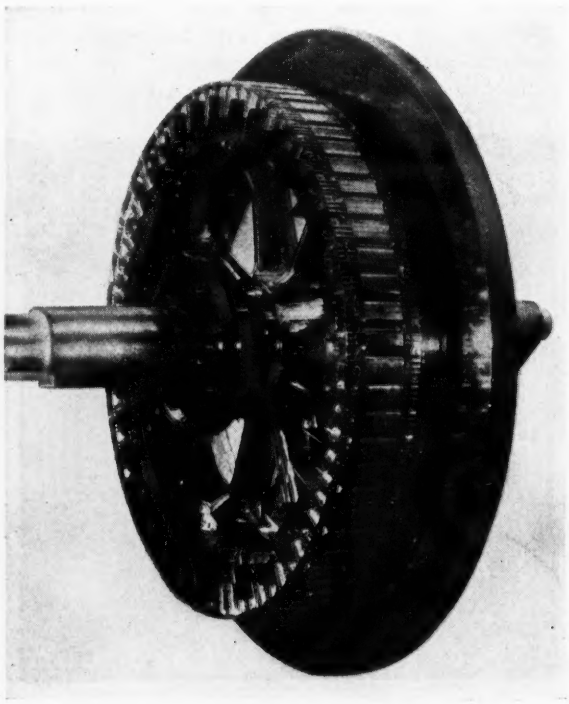

Fig. 7—Flywheel built as part of the rotor rather than a separately attachable ring

for adding extra WR^2 in flywheel rings, Fig. 8, which are disks that can be attached to the motor independently of the motor rating.

Inherent with this was the necessity of having motors of large diameters to be able to secure the necessary WR^2, and this meant narrow frames, a design permitting shorter distances between compressor frames and reducing the required floor space.

By having rings with different values of WR^2 and attaching one to each side of the rotor a large variety of WR^2 values can be obtained, so that almost any compressor operating condition can be handled without a separate flywheel.

Flywheel calculations have been still further simplified, for commercial compressors, by adopting com-

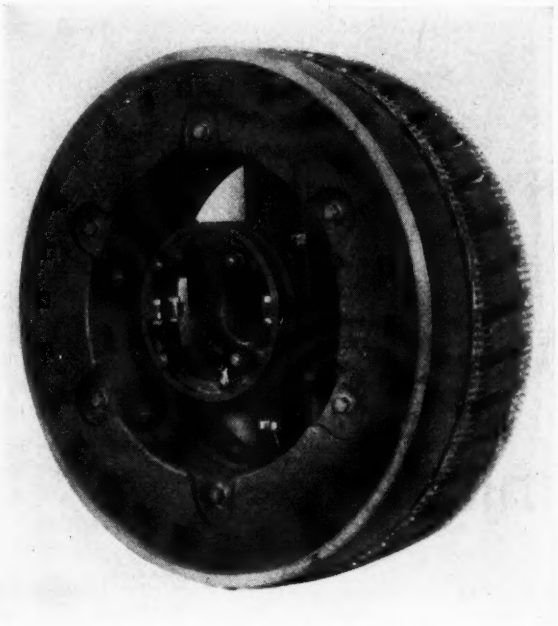

Fig. 8—Flywheel rings attached to the rotor to obtain full flywheel effect in the motor

pressor factors, designated as C, for each operating condition of each type of compressor. Each motor rating is given a motor factor F_c. The product of these two factors gives the required WR^2 directly. $WR^2 = C \times F_c$.

To determine the C values, or, more particularly, the XY curves from which they are taken, is a long, tedious process, but this method of calculating flywheels has proved very accurate, as shown by check tests as well as by lack of troubles.

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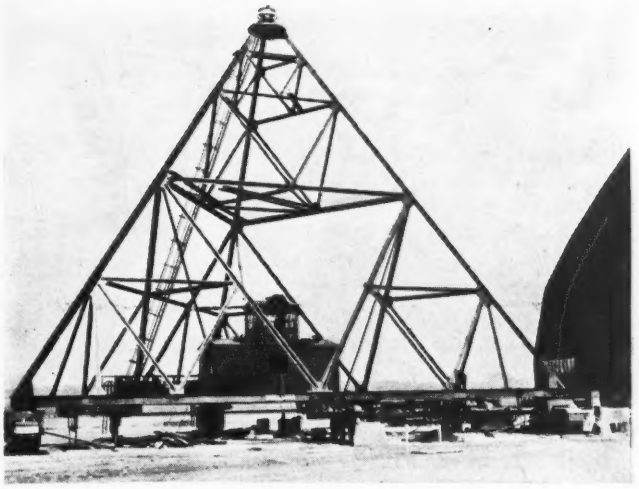
Akron's Mooring Mast Contains A 250-Hp. Power Plant

THE U.S.S. Akron while on the ground outside its hangar is attached to a three-legged mooring mast on caterpillar wheels, which is a recent development to facilitate the ground handling of large airships. This mooring mast is also used to tow the airship out of or into its dock and replaces the hundreds of men previously required to do this job.

The three legs of the tripod rest on caterpillar feet equally spaced on a 100-ft. circle. Two of the caterpillars provide the traction and are driven by a 125-hp., 250-volt direct-current motor mounted on the framework between them. The third caterpillar is for steering and is driven by an 8-hp. motor through a rack-and-pinion system.



The building at the center of the mast is for the power plant and the control room

The mast also carries its own power plant. Supported on the cross members of its base is a small house containing a 250-hp., 1,200-r.p.m. gasoline engine driving a 125-kw., 250-volt direct-current Westinghouse generator and a combination exciter and auxiliary generator rated at 18 kw., 125 volts.

Besides supplying power to steer and propel the mast, the main generator furnishes power for the 30-hp. winch motor, which hauls in the main mooring line of the airship. Simultaneous operation of the winch motor and the traction motor is prevented automatically by the contactor system. These contactors are mounted on a panel in the generator room. The mast, while serving as a mooring for an airship on the field, also supplies the power for servicing the ship with water and fuel. The motor-driven pumps are on the base cross members.

The 250-hp. gasoline engine drives a 125-kw., 250-volt direct-current generator to supply power to the motors for operating the mooring mast

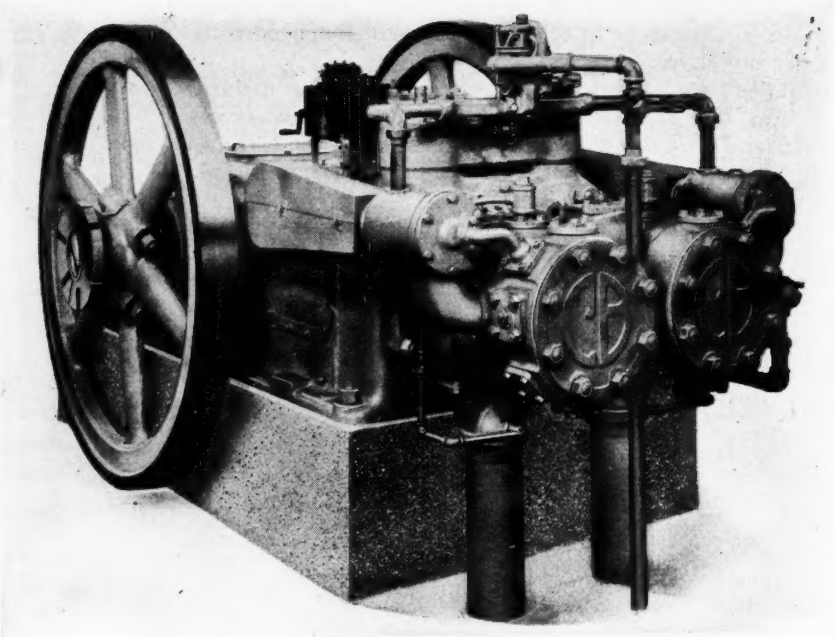


Fig. 1 — A twin-cylinder engine employing pump injection for the gas

By H. F. SHEPHERD

Two-Cycle Engines With Gas Injection

THE two-stroke-cycle gas engine, ranging from 10 to 200 hp. and used as the chief power source in oil field pumping and recompressing plants, lately has undergone a revolution through which its gas consumption has been made to equal that of the four-stroke-cycle engine. In accomplishing this marked improvement in engine performance designers have brought out various mechanical engine attachments. In place of scavenging the cylinder with an air-gas mixture, all employ the principle of clearing the cylinder with air only, introducing the gas fuel after the exhaust ports have been re-covered by the piston. All the modern gas-injection devices employed to increase the efficiency of two-stroke-cycle gas engines make use of the basic ideas incorporated in engine patents of 50 years ago, today's patents mainly dealing with the details by which these basic principles are made to function.

In the years following 1881 gas engine application had a vast growth in two fields where commercial gaseous fuel was distributed. In the cities gas engines replaced small steam engines, creating the small unit power demand that now belongs to the urban electric companies. In the oil fields gas engines replaced steam engines.

The cities were served with expensive gas of 500 to 650 B.t.u. heating value or with natural gas that grew costly as supplies were threatened and conservation for domestic use became the rule. The greater gas supplies had not yet been tapped and no enterprise had come forward to put the required millions into gas transportation systems.

The Southwestern oil fields had a great excess of gas over that required for power and no market for it. The conservation methods of today had not been applied. Consequently the industrial centers used four-stroke-cycle engines of the most economical type available, while the oil fields used the most primitive two-stroke-cycle engines. Mechanically the latter were sufficiently perfect to operate well over 99 per cent of the time each year. They were by no means as economical of fuel as the four-stroke-cycle engine, although far superior to the simple oil field boiler and steam engine, and there was no known need of conserving gas. The petroleum engineer did not know that a flowing oil well is a natural gas lift, the ascending column of fluid being lightened by formation gas and gas coming out of solution as the pressure on the oil is reduced, and he had not learned to pump this gas back into its natural reservoir to maintain the rock pressure, prolonging the natural flow of wells and conserving the gas for the future.

Oil fields with depleted gas pressures due to age or exploitation began to be short of gas. Whereas Eastern fields felt this after a generation of conservative production, Western fields, pushed to the limit to meet the demand for oil, were threatened with gas shortage in a few years.

The two-stroke-cycle engine had established itself as a reliable prime mover. It became desirable to better its fuel economy without the loss of its original virtues of low first cost, low upkeep cost and simplicity.

Its indifferent thermal efficiency was easy to account for. Its cycle was the same as the four-stroke-cycle Otto engine, and for the same compression quite as efficient. The negative work of pumping scavenging air might be much greater than the suction drop and exhaust back

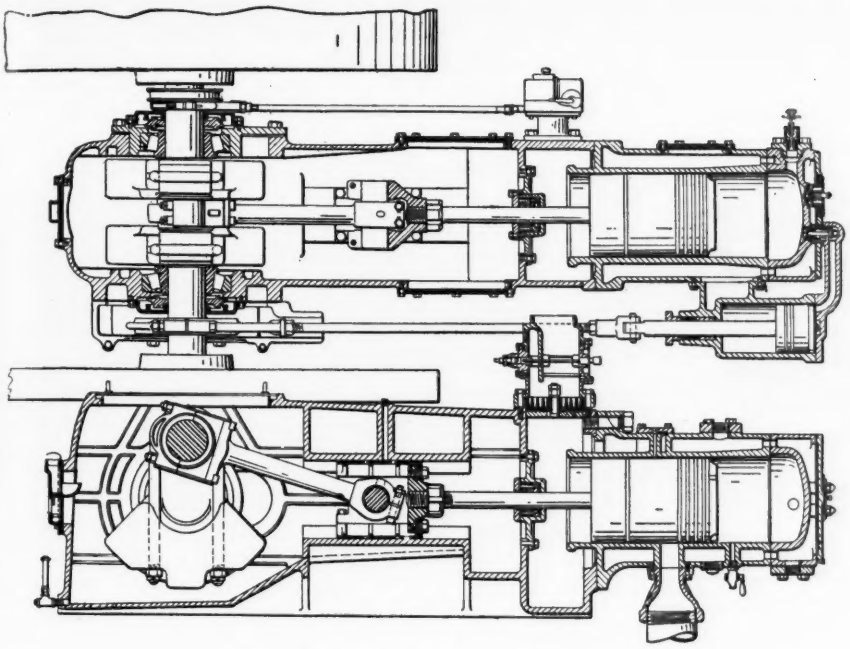


Fig. 2—Sections through a pump-injection engine

pressure losses of four-stroke-cycle engines, but not necessarily so. The loss was not due to inefficient combustion of the gas retained in the cylinder but to the gas lost through the exhaust ports while scavenging with the air-gas mixture. This fact was not new-born. It was a tradition. Every student of internal-combustion engine history knew the means of avoiding this loss, that is, scavenging with pure air, admitting gas afterward. The question was whether the oil field would adopt a new means of fueling gas engines, and various manufacturers produced devices whereby the gas was introduced after scavenging had ended.

These various devices are all based on patents issued in the 1880's.

The commercial success of the four-stroke-cycle Otto engine, which was patented in 1872, set men to prospecting for other methods of working without awaiting the expiration of the Otto patent. This led to a series of two-stroke-cycle engine inventions, beginning with the Alexander Rider patent No. 345,218 filed May 12, 1881, and patented Aug. 2, 1881. Claim 1 of this patent specifies:

"A gas engine provided with a motive cylinder constructed to exhaust the burned air, and air- and gas-supply pistons and cylinders, the parts being constructed and arranged to cause the charge of air entering the motive cylinder to rapidly expel the last portion of the air therefrom, after which the air is confined in the motive cylinder and the charge of gas then admitted after the exhaust opening has been closed, substantially as set forth."

Engines of this sort are classified by the late James T. Allen, in his Digest of U. S. Patents of Air, Gas and Calorie Engines, as follows:

"Subclass: Two-Cycle, Separating Air and Gas Pumps, two-cycle recompression internal-combustion engines in which air and combustible in the gaseous form are supplied to the working cylinder or combustion chamber by separate and distinct pumps, each operated by and in unison with the engine, one at least of the constituents being recompressed within the working cylinder before the ignition of the combustible mixture. The air is ordinarily supplied to the working cylinder of the engine before the gas, and such air is frequently designed to more or less completely scavenge the working cylinder. In such cases a part of the air thus supplied remains in the cylinder to form with the gas subsequently supplied thereto the next following charge. The air and gas may enter the working cylinder simultaneously. The air and gas are ordinarily intermixed within the working cylinder before the completion of the compresison-stroke, so that the complete charge is recompressed in the working cylinder before the ignition thereof and the beginning of the working stroke; but this subclass also includes engines in which one only of the constituents of the charge is recompressed in the working cylinder, in which case the other is supplied thereto under pressure during the working stroke."

From 1883 to date a number of patents were placed under this classification. The following is a copy of Claim 5 of the first Cyrus W. Baldwin patent, No. 276,750, filed Jan. 5, 1883, one of the twenty-two patents of this classification listed in Allen's Digest published in 1905.

"The combination of the cylinder, its exhaust port, piston, and air inlet valve and gas pump all arranged as set forth to permit the air to traverse the combustion chamber as the piston completes its forward

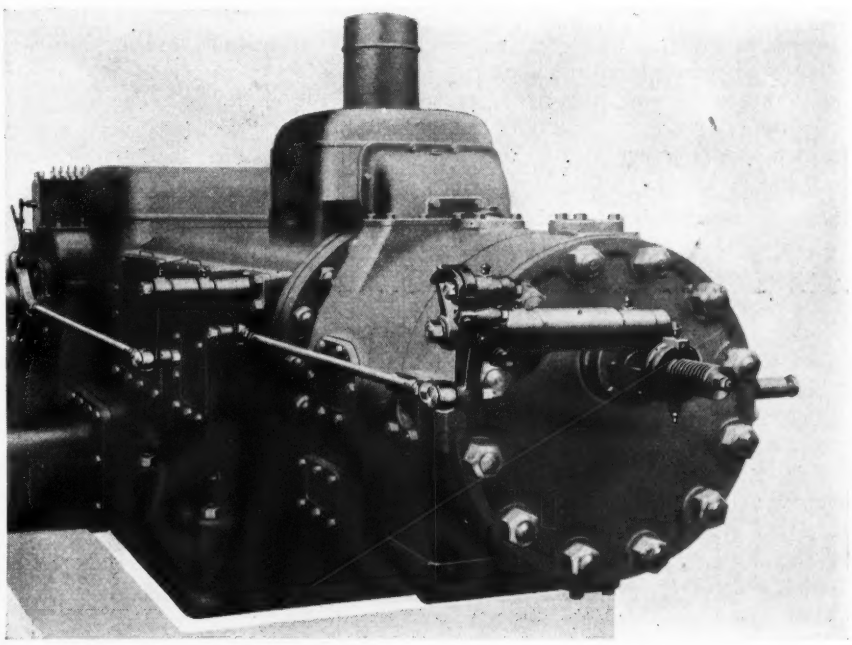


Fig. 3—An engine employing a mechanically operated gas-injection valve

movement to compress the air and force a measured charge of gas into the latter as the piston returns, and to compress the combined gases on such return movement of the piston substantially as set forth."

Except for valve scavenging, the engine shown by Baldwin on the drawing, Fig. 4, accompanying his patent

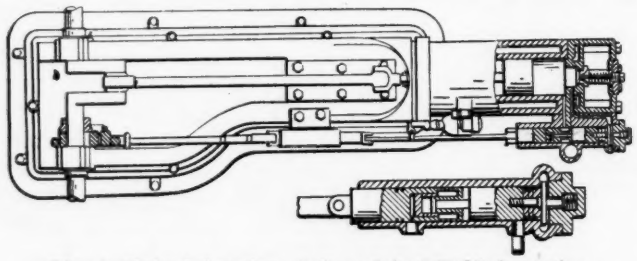


Fig. 4—Patent drawing of the original Baldwin engine

description and claims is strikingly like the most modern developments.

The patents allowed since are for structures and mechanisms intended to carry out the functions described. Naturally these later patents have longer and more detailed claims than the basic patents, limiting the scope of the inventions to specific combinations of parts not previously used, in the Patent Office examiners' knowledge.

Without doubt engines with the separate gas-charging pumps proposed by Rider and others would have met with a poor reception. They are available now, but few have been sold.

Fortunately, however, the larger oil field units were used for compressing gas, the compressor piston being connected to the engine crosshead by tierods. There was no reason why this compressor cylinder, which constituted the engine load, should not serve also in part as one of the "separate and distinct pumps . . . operated and in unison with the engine." The low stage, or, for that matter, the main discharge, could be bled, the gas passing to the engine cylinder through a reducing valve. This, in fact, was done, and today the method is in broad commercial use. Engines for belt or electrical transmission of power are provided with directly driven or separately driven fuel compression units.

The mechanism for admitting fuel to the engine cylinder after scavenging is simple. To date it is a simple inward opening poppet valve, mechanically operated when gas is taken from the mains of a compressor station and automatically or pressure-operated when the engine is equipped with an injection pump. A valve of the former sort is shown in Fig. 3, while an injection pump and valve appear in Figs. 1 and 2. Governing is by regulation of gas pressure at a simple throttle.

It is natural that engines of this type should have special properties to be understood and mastered by operators, although no insuperable difficulties have been encountered.

The oil field compressor plant operates under conditions which never can be predicted or defined. Changes of intake pressure may throw the compression ratio toward or away from the critical ratio of maximum power requirement. Compressor cylinder sizes are changed at need, but it is economical to operate existing equipment as long as possible.

While intake pressures are specified in the compressor engine contract, operators desirous of putting as much gas through the plant as possible raise the suction a few inches or a few pounds, keeping just below the distress load on the engine. Perhaps 25 lb. more head pressure will overcome pipe line resistance and shove more gas through. No blame attaches to this. No job has more risks than oil field compression plant design. Conditions are only a matter of conjecture until the

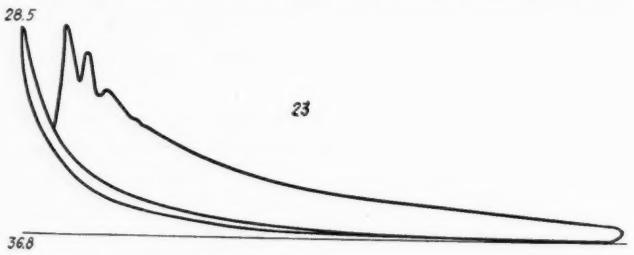


Fig. 5—Result of a poorly mixed gas and air charge

gas supply is drawn upon, and even engineering management is not above asking why.

In short, overload is the rule in compression plants. Consider a compressor operating at 200 r.p.m. and fully loaded, that is, pulling its load with a wide-open governor valve. The mixture being burned in the engine cylinders is of maximum richness. The gas admission valve is open during 30 deg. of crank travel, or $\frac{30}{360} \times \frac{60}{200} = \frac{1}{40}$ sec. A slight change of conditions adds a little load to the unit, which has been operating under exact equilibrium between its maximum power output and its load, and the unit slows down.

The fuel-gas flow is being governed by the pressure and valve orifice area, together with the time duration of opening. Slowing the engine extends the time of flow.

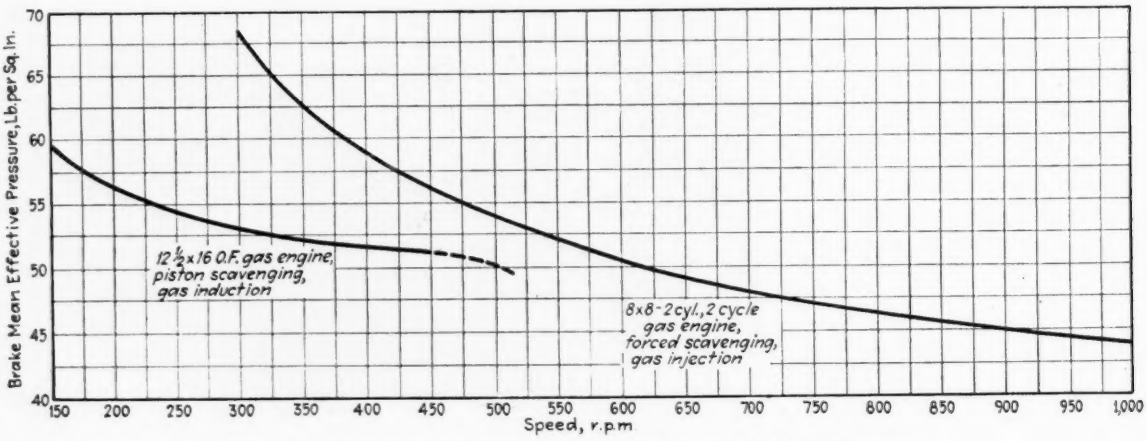


Fig. 6—Comparative pressures obtained with induced and injected gas

The mixture richness is increased beyond ignitability and the engine stalls.

An engine taking fuel by induction is more stable. Gas is drawn in by the pressure drop in the intake during aspiration. If the engine slows, the intake manifold pressure rises and less gas is drawn. It may drag down and hang on. Not so the injection engine drawing gas from a constant-pressure system. Its draft of gas is a function of orifice area and time. If the engine slows under maximum load it can do nothing else but stall, since the time of gas admission is increased and the mixture is made too rich to fire. Engines of this class should be operated with a 10 per cent power reserve to insure perfect reliability.

The commercial engine fueled by injection of gas into the air charge has nothing in common with the diesel engine. Gas is admitted as early as possible in the compression stroke to insure thorough mixing before ignition. Mixing is not accomplished by injection, for the entering gas tends to blow the air out of the way and remain in a body. The natural turbulence of both the air charge and the gas effects the mixture

Usually gas is admitted during 30 deg. of crank travel; theoretically the point of admission coincides with exhaust port closure. This is a good point to work from if the conditions of exhausting and scavenging are not well known.

At higher speeds, 400 to 1,000 r.p.m., the time between completion of injection and ignition may not suffice for mixing. It then becomes desirable to advance the point of gas admission. No appreciable loss results if the admission coincides with inlet port closure, and carefully designed exhaust systems may make possible a still greater advance without loss.

There is no difficulty in arriving at a correct diagnosis of a case of incomplete mixture at firing time. If mixing is not fairly complete the charge simply fails to fire or firing is erratic, producing diagrams like Fig. 5. A little manipulation of the ignition time shows a sharply defined angle at which ignition is regular. A few degrees advance from this position may result in complete failure to fire. If this firing angle is too near dead center to produce well-developed power diagrams the injection must be advanced to allow time for mixing. Fortunately, this action is evidenced only at speeds well above the requirements of any oil field compressor engine.

Most oil field engines use the displacement of the crank end of the power piston for pumping the scavenging medium, which until the advent of the injection engine was the fuel-air mixture. The relation of power cylinder displacement to scavenging pump displacement is, then, one to one. It is well known that thorough scavenging is not possible with this relative volume. Diesel engines use approximately one and a half power cylinder volumes of air for scavenging. If this is attempted in gas engines using the air-gas mixture for scavenging the spill at the ports will cause great fuel loss, since the cylinder cannot possibly contain more than its own volume of mixture.

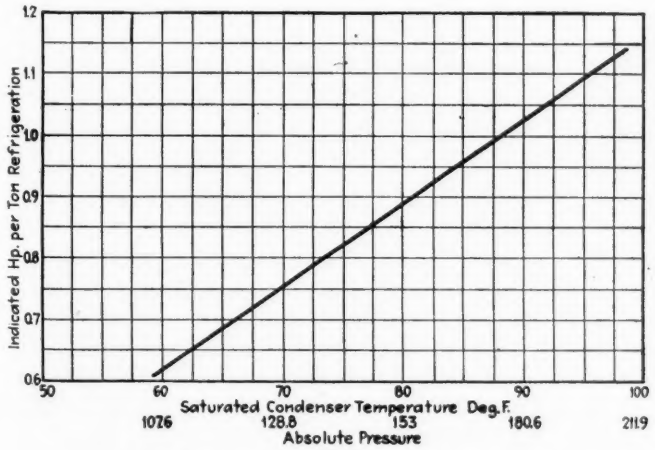
When two-stroke-cycle engines are fueled by the injection method air scavenging of the same efficiency as that used for diesel engines may be employed without fuel loss and with much increased output, as shown in Fig. 6.

▼ ▼ ▼

The Value of Clean Condenser Surfaces

By M. F. KNOY

PERHAPS there is no other factor about the ice plant that has a more direct bearing on power consumption than the condition of the heat-transfer surfaces of the ammonia condenser. Whatever type of condenser is used, some means of cleaning the water side of these surfaces must be employed. The degree of difficulty of this process and the length of intervals at which it is required will depend largely upon the nature of the cooling water. Some waters will deposit a hard, tenacious scale, some only a soft coat of slime, others only silt and mud, and others practically nothing.



Variation of horsepower with condenser pressure

The effects of fouled condenser surfaces will be apparent in the form of high head pressures. A coat of foreign matter on these surfaces retards the transfer of heat from the ammonia to the cooling water. This requires a greater mean temperature difference between the two fluids, which can only be attained by a corresponding rise in the condensing temperature of the ammonia. This, of course, results in increased head pressure. The higher head pressure not only increases the mean effective pressure in the compressor, but it lowers the compressor's volumetric efficiency as well. It thus serves to increase the power consumption per ton of refrigeration by both of these means. The graph shows how power requirements per ton of refrigeration increase with the increase in head pressure. In this case the evaporator temperature is assumed to remain constant at 5 deg. F., which corresponds to a suction pressure of 19.6 lb. gage.

The nature of the cooling water should be taken into account in specifying the type of condenser to be used. If frequent fouling is expected, the type of condenser most easily kept clean under these conditions should be selected. Spacing should be such that plenty of room for cleaning operations is provided.

PRACTICAL AIDS TO OPERATION

How the Pump Motor was Kept in Service

WHILE EMPLOYED as maintenance man at an isolated power plant an operator of one of the process pumps reported that the motor had stalled. This was a 5-hp. unit with roller bearings. Everything checked all right, including the rotor. The bearings were practically worn out, but the rotor was not rubbing the stator. As the pump was needed badly and it would be several days before new bearings could be obtained, a temporary repair had to be made. Taking up wear in these bearings is accomplished by removing laminated shims between the faces of the bearing housing and the end bell of the motor. It so happened that all of these shims had been removed at some previous time. All of the take-up provided by the bearing manufacturer had been used. Being sure that the bearings were at fault, a washer was cut out of No. 20 gage iron the same size as the bearing thrust. It was placed against the bearing and the housing bolted up tight. The motor was started and operated satisfactorily.

If a more permanent repair had been desired the bearing housing could have been turned off about $\frac{1}{16}$ in. on the finished face. This would have answered the same purpose as the washer. Of course, when new bearings were installed it would have been necessary to shim up for the amount of metal removed.

This method would not be recommended for a permanent repair.

G. H. EMERSON.

Tarrant, Ala.

Seal Pipe on Feed-Water Heater Solves Overflow Difficulty

AN OPEN-TYPE feed-water heater was furnished with an overflow device consisting of a balanced valve actuated by a float incased in an external chamber set at the maximum allowable water level in the heater. This device was in communication with the heater through a vertical pipe running down to a connection at about the heater's midheight, also through an equalizing pipe extending from the float chamber to the top of the heater. A float inside the heater actuated a regulating valve in the water-supply pipe. Exhaust steam was supplied at a pressure of about 8 lb., which kept the feed water at a temperature close to 235 degrees.

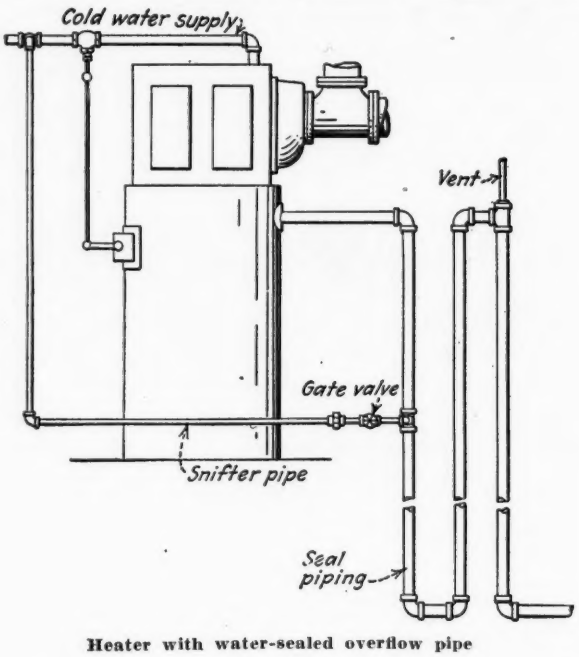
It happened occasionally that friction in the joints and stuffing box would temporarily prevent movement of the float mechanism, so that the valve in the water-supply pipe would remain closed when it should have been open. Then when the water level had fallen to a low point the resistance would suddenly vanish and the supply valve would pop wide open. The instant effect would be a great inrush of cold water and the creation of a partial vacuum. Acting through the equalizing pipe, the vacuum would pull the water already in the heater up into the overflow chamber and there would be a great outrush through the overflow pipe.

Occasionally, also, a contrary action would take place —the float mechanism sticking while the valve in the supply pipe was open. Flooding of the heater would result.

To avoid these difficulties the operating man decided to cut a hole in the shell of the heater at the maximum water level and attach a simple overflow pipe with a seal in it, as shown in the accompanying sketch. He figured that by dropping the loop in the piping to a depth of 22 ft. he would have a column of water sufficient to seal against the 8-lb. steam pressure in the heater.

But when first tried the device did not work as anticipated; all the water was blown out of the piping and was followed by a blast of low-pressure steam. That was because no provision had been made for giving the descending leg a "snifter" of cold water. After this omission had been supplied, as indicated in the sketch, the vent worked satisfactorily.

The cold water was needed because of the comparatively high temperature of the heater water. At the instant the overflow ceased, the ascending leg of the piping was filled with water at a temperature of about 230 deg. and under atmospheric pressure through the vent



Heater with water-sealed overflow pipe

pipe. Consequently, this water immediately flashed into steam and away went the seal. By means of the cold water the temperature of the overflowing water could be reduced below the boiling point under atmospheric pressure. Thus a complete column of water could be retained in the ascending leg and a column not less than 2 ft. high in the descending leg.

But admitting the cold water by means of a hand-operated valve just when it was needed called for greater vigilence on the part of the boiler room crew than could reasonably be expected. So the device was made automatic by installing a thermostatically controlled valve in place of an ordinary gate valve, the thermostat being attached near the lower end of the descending leg.

St. Louis, Mo.

D. J. ALLISON.

CURRENT COMMENT

Operating Characteristics of Overfeed Stoker

IN THE Dec. 15 number the article on "Operating Characteristics of Overfeed Stokers" contains the statement: "In general it may be said the overfeed type is best suited to the free-burning and low-volatile coking bituminous coals found in the Eastern States, which require some agitation of the fuel bed and are not particularly difficult to ignite." The Chicago Automatic stoker therein mentioned can burn any kind of coal regardless of its ash or sulphur content. With coal containing a great amount of sulphur and ash one does not dare to agitate it, because clinkers form rapidly. This makes it difficult for an overfeed stoker with moving grate parts to handle this kind of coal. It clinkers so rapidly it will not carry the load. The Chicago stoker is a non-agitating type. There are no moving parts under the fire, therefore it can burn coal that will not permit agitation, and the cheaper coals must not be agitated.

The same paragraph says: "Some types of overfeeds can handle a much wider range of fuels than others, depending largely upon the amount of agitation that can be given to the fuel bed." This is the very thing they must not do. They must not agitate the cheaper grades of fuel. The Chicago stoker spreads the coal evenly over the entire grate area, by means of the various angles of the blades, which are adjusted when the stoker is installed, and they throw the coal to all parts of the grate surface. This is done from the outside of the boiler, 14 in. above the fire. A fuel bed as thin as one inch in thickness is carried.

You further list this stoker as a forced-draft stoker, whereas it never uses forced draft. The feed is continuous and the fine coal is burned in suspension and the heavier coal drops thinly and loosely on the grate surface. It is so thinly and loosely spread that forced draft is never necessary regardless of the load conditions.

Chicago, Ill. N. T. LADENSON,
President
Chicago Automatic Stoker Corp.

Surface Tension and Priming

IN THE REPLY to the question "What is surface tension?" published in the Dec. 29, 1931, number it is stated that "surface tension causes foaming." Should not there have been included in the statement that priming is the consequence of excessive foaming induced by surface tension?

A new two-pipe gravity steam-heating system during its first week of operation gave considerable alarm to the operator due to the boiler suddenly losing its water whenever the pressure was raised above 2 lb. gage. The boiler was of the horizontal return-tubular, hand-fired, all-welded type and the manufacturers apparently had painted the shell interior with some oily preparation to prevent active corrosion while in stock. On being notified of the trouble in keeping water showing in the gage glass the somewhat dubious contractor ran a test to see what was the trouble. It was found that conditions were satisfactory and that the water level remained as manually controlled until the pressure was raised to 2 lb. gage and over, whereupon the water appeared to drop out of sight in the glass. A thin film of oil showing on the top of the water indicated the source of trouble.

The safety valve mounting was taken off the shell and a pipe connection run to the sewer after which 10 lb. of soda was mixed in a bucket of water and dropped into the boiler water. The water level was then run up to the top of the shell and a strong fire was kept burning until the interior was thoroughly boiled out, care being taken to maintain the water level. This was found to cure the trouble, which had evidently been the direct result of reduced surface tension caused by the action of impurities on the water level.

Another case occurred with a 300-hp. water-tube boiler using pulverized coke breeze as the fuel. In this case the boiler operator reported he could not maintain the level in the glass with the feed-pump flow-meter indicator around to its limit, that is, at full rating, even with the boiler idling (at 80 lb. gage). The trouble was located in the water hardness. In this plant it was the operating practice to take out about 10 in. of salt brine after 16,500 gal. has passed the zeolite softener to the feed-water heater. Instead of the 10 in. of salt brine being taken out, the whole brine tank had been emptied at one regeneration, due probably to the operator's attention being called to trouble on a near-by compressor, with the consequence that the boiler-water under notice became too salt. The remedy adopted, as the other boilers were under repair and not available for cutting in at the moment, was to bypass the softener and blow down the boiler at intervals until normal hardness was reached. This corrected what was evidently priming caused by increased surface tension due to salt.

Priming in both these cases was not caused by either insufficient steam space or by forcing the boiler. Accounts of other instances of priming and the remedy adopted would be interesting. LESLIE W. ROSS.

Toronto, Ont.

High Temperature Causes Fuses to Blow

IN ELECTRICAL installations proper attention is frequently not given the location of fuse blocks in regard to temperature. A speed-control box for an automatic stoker in one plant was mounted against the brickwork on the side of the furnace wall. The main leads to the stoker motor ran through fuses inside of this box. Some time after the plant was placed in operation and the boiler was operating at a high rating the fuses blew out several times. It was apparent that something was wrong.

The electrical connections were checked and found not to be at fault. Then the engineer discovered that the temperature of the fuse box was around 120 deg. F. due to the heat conduction through the furnace wall. This temperature was sufficient to cause the fuses to open at full load. The control box was removed and several sheets of insulating material were put between it and the wall, after which there was no further trouble.

Milford, N. J. HARRY M. SPRING.

READERS' PROBLEMS

PUMPING WORK—*How does one calculate the work done by a pump in handling water?* C. C.

Work is defined as force times distance. The work done in pumping water is the weight of water pumped multiplied by the vertical distance (or its equivalent) that the water is raised. For example, the work done in raising 1,200 lb. of water through a vertical distance of 50 ft. is $1{,}200 \times 50 = 60{,}000$ ft.-lb. Pumping water against a suction or discharge pressure is equivalent to lifting it against gravity, each pound per square inch of pressure or suction counting as 2.3 ft. of lift, or "head."

Suppose that a gage at the pump discharge shows a pressure of 140 lb. and that the center of this gage is 8 ft. above the water level in an open tank from which water is being pumped. Then the total "head" pumped against (not counting the friction in the suction pipe) is $8 + 140 \times 2.3 = 8 + 322 = 330$ feet.

SCALLOPED BOILER SEAMS—*One of our high-pressure steam-jacketed tanks has the seams provided with coverplates, the outside one having the edges scalloped. What is the reason for this?* M.P.

The serrated, or scalloped, edge is provided to reduce the length of calking edge between rivets. Examination of the joint will show that the distance between the rivets of the outside row is greater than the distance to a rivet on the next row.

POLARITY OF INTERPOLES—*What is the relation between the polarity of the interpoles and that of the main poles in a direct-current machine? If a four-pole machine has only two interpoles what polarity do they have?* R.E.S.

A simple rule for remembering the polarity of interpoles is that they have the same polarity as the armature's magnetic poles under them, that is, a north armature magnetic pole should have a north interpole over it. In a motor the armature will rotate in a direction such that the magnetic poles of the armature will tend to move away from the main poles that have a polarity similar to those of the armature. In other words, the north and the south poles of the armature will tend to move away from the north and the south main poles, respectively, and toward the south and the north poles. From the foregoing it is evident that on a motor the polarity of an interpole is the same as that of the main pole behind it in relation to the armature's direction of rotation.

On a generator the armature is rotated in a direction tending to move its magnetic poles toward the main poles having the same polarity as the armature. Since the polarity of the interpoles is the same as that of the armature, their polarity will be the same as that of the main poles just in front of them, relative to the armature's direction of rotation. By remembering that the interpoles have the same polarity as the armature and that the armature's magnetic poles on a motor tend to move away from the main poles having a similar polarity and toward these poles in a generator it is easy to picture the relation between the interpole and main pole polarity.

When there are only two interpoles in a four-pole machine they will have the same polarity and may be either two north or two south poles, depending upon how the machine is connected and its direction of rotation.

DECLINE IN COSTS OF DIESEL ENGINES—*What has been the change, if any, in the sales price of diesel engines in recent years?* G. A.

The cost of any individual engine will vary with the manufacturer and with the accessories supplied. In case of 400-hp. four-stroke-cycle diesels of one make the costs f.o.b. factory for the engines arranged for oil pipe line work, that is, without generator but with the usual accessories, were:

Year	Cost per Hp.
1924	$71.00
1924	65.70
1925	56.33
1926	55.40
1927	50.60
1928	50.69
1929	44.90
1930	43.49
1931	41.50
1931	34.56

RADIATOR PAINTS—*Is there any disadvantage in painting radiators, and if so what is the loss in heating effect?* N. C.

Calling the heat emissibility of a bare cast-iron radiator 100 per cent, the heating efficiency with various paints is as follows:

Copper bronze	76
Aluminum bronze	75
White enamel	101
Green enamel	95
Terra cotta enamel	103
White lead paint	99
White zinc paint	101

▼ ▼ ▼

Balancing Refrigerator Compressor Load

THE QUESTION

*IN OUR PLANT we have a double-acting twin 5 x 6-in. compressor, running at 190 r.p.m. * * * The receiver is 6½ in. x 6 ft.; the brine tank is 3.25 ft. x 8 ft. x 4 ft., with one 252-ft. coil of 1¼-in. pipe. * * *

We have four rooms; the dimensions are: No. 1 room 570 cu.ft., in which the temperature is kept at 18 deg., with 220 ft. of 1¼ in. pipe; No. 2 room is kept at 30 deg. with 150 ft. of 1¼-in. pipe and has 190 cu.ft.; No. 3 room is 380 cu.ft., the temperature being kept at 40 deg., with 90 ft. of 1¼-in. pipe; No. 4 room is 343 cu.ft., the temperature being kept at 40 deg., with 75 ft. of 1¼-in. pipe. The three expansion valves are placed in a header near the compressor, one for the tank, one for the No. 1 and 2 rooms and one for No. 3 and 4 rooms.*

We cannot get the frost back to the machine from Room 4. If we open the expansion valve wider we get the frost back from Room 4, but we do not get frost from Room 2. Only a few feet from one to the other frost; the rest of the pipe remains wet. G.M.

IT HAS always been a mystery to the editor how a radio message from one ship to another permits a doctor to prescribe for the seamen on the doctorless ship. Unfortunately the prescription only is made public, the patient's final reaction remaining shrouded in doubt. In answering this particular refrigeration question the engineering doctors fail to agree in their long-distance diagnosis, but all contain certain truths which if seized upon may correct the trouble. Let us compare the prescriptions.

GIVING a complete diagnosis, M. F. Knoy, Beaumont, Texas, states that often in refrigeration literature one sees the injunction "Always keep the suction line frosted back to the compressor." This is unquestionably a good rule to follow under normal conditions when the compressor is working on a single evaporator. But when the compressor is working on two or more evaporators in parallel it is frequently impossible to maintain frost on each of the suction lines.

The condition reported is not faulty operation but is due to the fact that there are several evaporators in parallel variously loaded and serving compartments at various temperatures. For example, Room 4 is held at a temperature of 40 deg. F. If we assume a practically constant rate of heat re-

moval from the room it means that a certain arithmetical mean temperature difference must be maintained between the air in the room and the refrigerant in the coils. And since the initial temperature of the refrigerant must remain at approximately —1 deg. F. to correspond with the 15-lb. suction pressure, the final temperature of the refrigerant at the coil outlet will be determined by the mean temperature difference maintained.

To be more concrete, let us assume that the refrigerating load on Room 4 requires an arithmetical mean temperature difference of 22 deg. between the air in the room and the refrigerant in the coil, and that the room is held at 40 deg. The average temperature of the refrigerant will then have to be 40 — 22 = 18 deg. Since the initial temperature of the refrigerant is —1 deg., the final temperature will be 38 deg. This temperature is, of course, too high to give frost on the coil outlet. On the other hand, Mr. Knoy continues, if we assume a load increase in Room 4 such as will require a mean temperature difference of 30 deg. to effect the necessary heat removal, the average temperature of the refrigerant will be 40 — 30 = 10 deg. The initial temperature of the refrigerant will, of course, remain at —1 deg. to correspond to the suction pressure, and the final temperature of the refrigerant will be approximately 21 deg. In this case the coil outlet would be frosted.

It is seen that when several evaporators are operating in parallel on the same fixed suction pressure the temperature of the outlet gas from each coil will be determined separately by the air temperature and the refrigerating load in its respective compartment. The operator will therefore be unable to control the frosting back of the suction lines.

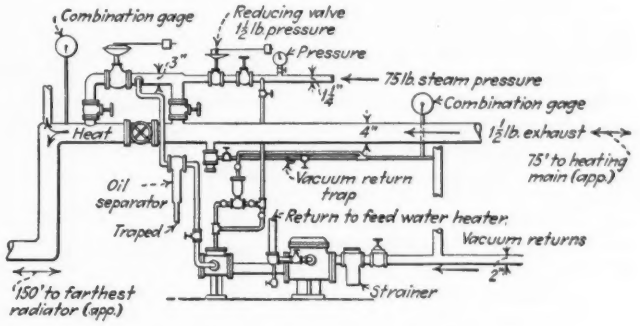
Unless circumstances warrant the installation of a dual-suction pressure system of some kind, the questioner must be satisfied to maintain the desired temperatures in the various compartments and let it go at that.

A Question for Our Readers

IN THE ILLUSTRATION is a sketch of a problem in heating. Steam is being used from an intermediate steam pressure, although originally the heating was to be done by exhaust steam, especially in mild weather.

The problem is—Why does this present hook-up not work when the valves are changed to use exhaust steam? As soon as the change is made the vacuum pump loses its vacuum, 8 to 9 in. and begins to run a very undesirable speed. There are two combination pressure and vacuum gages; A is connected to the main heating line and B is connected to the vacuum return; A may have 1½ lb. pressure from either the exhaust or from the intermediate pressure reduced to 1½ lb. pressure. As soon as exhaust steam is used B loses its vacuum and in a short period of time will read the same as A.

R.E.F.



Arrangement of valves on heating system

Suitable answers from readers, if received promptly, will be paid for when published.

THE LENGTH of the coils in the separate rooms are at fault, in the opinion of August Ulmann, Jr., Norwood, Pa. His diagnosis starts with the premises that in the statement of the problem nothing is said to indicate whether the room temperatures are maintained as desired. If it is difficult to keep the rooms at the desired temperatures, a somewhat larger machine is required.

If it is possible to keep the room temperatures as desired, the fact that no frost appears on the vapor pipe from Room 4 shows that the coils in Rooms 3 and 4 are a little too large for the load imposed by the rooms and their contents. As the temperature of 40 deg. at which these rooms are kept is 8 deg. above freezing, it is not at all unlikely that the return gas will be above 32 deg. if the coils are too large, and thus no frost will appear on the line. The absence of frost on the suction line is only significant, declares Mr. Ulmann, when the room temperature desired cannot be maintained. In other words, under these conditions it is an indication of overload on the compressor.

The absence of frost on the line mentioned will not be detrimental to the performance of the compressor unless the line is long, in which case the returning gas will be superheated and its density so lowered that the compressor will get a greatly reduced weight of gas on each stroke. It must be borne in mind that the suction line is nothing more than a continuation of the cooling coils inside the rooms. For this reason these lines should be efficiently insulated, as any heat absorbed by the gas in transit between the cool-rooms and the compressor is a loss.

THE REMEDY is to have an individual expansion valve for each room, according to Fred MacDonald, Sioux City, Iowa. His explanation is that when the engineer opens the expansion valve wider on Room 4 the suction on the return gas is stronger and causes Room 2 gas to warm up and so give a wet pipe, because the temperature of the room is high. The only way to overcome that is to have a separate expansion valve, and a separate coil for each room, tapped into the main suction return header line.

IN THIS he has the support of A. B. Conklin, Baldwin, N. Y., who points out that for closer regulation there should be an expansion valve for each box which carries a different temperature. The coils in the box with the higher temperature will have a higher pressure and this will tend to choke those coils. Where No. 3 and 4 valves are opened wider there is a lack of liquid to No. 2 box, which is at a higher temperature than No. 1. Lack of liquid ammonia in the system would aggravate the condition and, as there is no complaint of flooding from box No. 1 and as the suction pressure is a trifle low, he suggests checking the amount of liquid.

For the best results under the existing conditions a compressor with a high- and low-pressure suction and some method of controlling the capacity is recommended. This would increase the capacity of the whole system by allowing a much higher suction pressure in boxes No. 2, 3 and 4.

AN ENTIRELY different diagnosis is made by John G. Fairfield, Troy, N. Y., who says that the symptoms indicate not enough ammonia in the system. This suspicion is borne out by making computations on the receiver capacity. This machine, having a piston displacement of 52 cu.ft. per minute, with an assumed volumetric efficiency of 85 per cent, and allowing about 4.5 cu.ft. per ton of refrigeration, must be rated at about 11 tons. If one allows 120 ft. of 1¼-in. pipe per ton, the total of 787 lin. ft. gives less than 7 tons of refrigeration. For average conditions, about 0.5 cu.ft. of receiver capacity should be allowed per ton; this means between 3 and 6 cu.ft. of receiver capacity in this system, but only $\frac{3.14\ (6.5)^2 \times 6}{4 \times 144} =$ 1.38 cu. ft. is installed.

A machine of this size should have around 250 lb. of ammonia in the system; but even if this receiver were full, it could contain only 80 lb. of liquid. The remedy consists in installing a larger liquid receiver and a suitable charge of ammonia; and try purging the dead gases.

Calculating Steam and Power Costs In Industrial Plants

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THE PROBLEM of determining the unit costs for energy and heat is a breaking-down process made possible only by adequate, reliable records. These records should be correlated to provide the best mental picture, as for example the flow sheet. Such records are attained only through the installation of sufficient meters, the cost of which, it will invariably be found, is meager compared with the savings made possible by the information they disclose.

The calculations which follow show how this breaking-down process can be applied to an industrial plant to determine the cost of power and steam per unit of product.

The quantities of electrical energy and steam required per unit of product manufactured have purposely been made fictitious, to simplify the problems and to prevent industrial managements reading this article from striving to parallel the energy costs.

Before considering the complex conditions in an industrial power plant, a simple case of a boiler that delivers 20,000 lb. per hour of 150-lb. saturated steam to a process S through a reducing valve at 6 lb. gage will be worked out.

The total heat of 150 lb. saturated steam above 32 deg. F. is 1,195 B.t.u. per pound, and, for brevity, assuming no radiation loss, each pound of steam leaving the reducing valve serving process S must have the same total heat at 6 lb. gage, and be superheated 79 deg. F. to a total temperature of 309 deg. F. process S is served with 20,000 $\times$ 1,195 = 23,900,000 B.t.u. per hour.

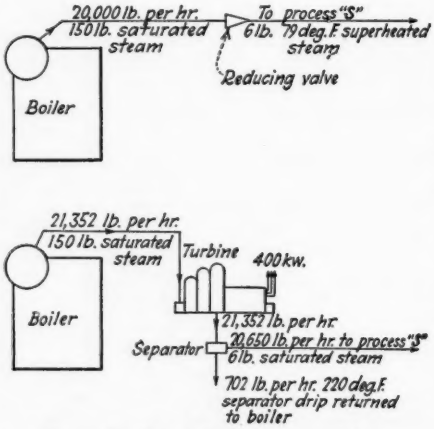
If, however, process S is served with 23,900,000 B.t.u. per hour from a noncondensing turbine or engine (which may be *likened to and replace the reducing valve)* exhausting at 6 lb. gage, and inserted in the steam line between the boiler and process S, a certain amount of heat is extracted from the steam through converting heat energy into motive and electrical energy. This heat is equivalent to about 3,710 B.t.u. per kilowatt-hour generated or 2,770 B.t.u. per brake-horsepower developed. This figure of 3,710 B.t.u. per kilowatt-hour is elaborated on later.

When serving process S with 23,900,000 B.t.u. per hour, it may be found that the turbine would generate 400 kw. To generate 400 kw. the turbine will extract from the steam fed to it heat amounting to 400 $\times$ 3,710 = 1,484,000 B.t.u. per hour.

It is possible with a turbine having a very poor thermal efficiency, if receiving 150 lb. gage saturated dry steam, to exhaust steam slightly superheated. In this case, however, as customarily, the exhaust steam will be wet. The separator will extract the moisture and, if 100 per cent effective, it will discharge dry saturated steam having a heat content of 1,157 B.t.u. per pound to process S.

The amount of steam required for process S will be $\frac{23,900,000}{1,157} = 20,650$ lb. per hour.

The moisture in the exhaust steam will be collected in the separator and discharged as water at 220 deg. F. and having a total heat of 220 — 32 = 188 B.t.u. per pound.



Two simple methods of supplying steam to a manufacturing process

Let X = the pounds per hour of moisture as water discharged from the separator. Then the steam fed to the turbine must be X + 20,650 lb. The total heat content of the steam fed to the turbine must equal the heat extracted by the turbine in generating 400 kw. plus the total heat of the saturated dry steam going to process S plus the total heat of the separator discharged water, or $(X + 20,650) \times 1,195 = 1,484,000 + 23,900,000 + 188\ X$.

Solving this equation, we find $X = \frac{23,900,000 + 1,484,000 - (20,650 \times 1,195)}{1,195 - 188}$

$= \frac{707,150}{1,007} = 702$ lb. per hour.

With 702 lb. of water per hour entrained in the turbine exhaust, the separator drip will amount to $\frac{702}{20,650} =$ 3.39 per cent.

The steam fed to the turbine will be 20,650 + 702 = 21,352 lb. per hour. This latter quantity must also be the boiler evaporation, whereas if no turbine were used then process S would properly be cared for with a boiler evaporation of 20,000 lb. per hour. The steam rate of the turbine will be $\frac{21,352}{400} = 53.4$ lb. per kilowatt-hour.

It should be emphasized that the relative amount of heat actually used by the turbine is $\frac{1,484,000}{21,352 \times 1,195} = 5.8$ per cent of the total heat content of the steam going into the turbine.

Often, non-condensing turbines and engines are erroneously charged with as much as 50 per cent of the steam as heat that enters the turbine. This, naturally, reflects against he power plant, resulting in the industrial management's believing its mill products, charged with the remainder of heat for processing, to be produced at an extremely low unit heat rate, although it is unable to market its products, with a fair profit, as cheaply as their more modern competitor.

It is notable that the heat cost of generating electrical energy as outlined above, if the small plant has a 75 per cent boiler efficiency, when burning fuel oil of 6,200,000 B.t.u. per barrel at 90c. per barrel, is $\frac{3,710 \times 0.90}{6,200,000 \times 0.75} =$ $0.000719 per kilowatt-hour.

This is 0.7 mills per kilowatt-hour for heat only, or about one-fourteenth of what it would cost to purchase electrical energy of a like quantity. It should be noted that fixed charges and other operating charges should be added to this to get the true cost.

Fixed charges on the additional investment necessary due to power generation, over the cost of a steam plant to deliver steam only at a maximum of 150 lb. per square inch, would be about 1.0 mill. The total comparative cost would be 1.0 + 0.7 = 1.7 mill per kilowatt-hour, or about one-sixth of what it would cost to purchase a like quantity of power.

To bring out some of the detail involved in determining the flow sheet data for an industrial manufacturing certain products given below, one of numerous possible power plant schemes,

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that of 425 lb. pressure 700 deg. isolated power plant, will be considered. The products to be manufactured are:

Product A: To be produced at the rate of 2.5 tons per hour and requiring 20,000 lb. per hour of saturated steam at 40 lb. gage and using 600 kw. of electrical energy. This might be typical, but as to method of use only, of rotary straw digestors used in paper mills.

Product B: To be produced at the rate of 4 tons per hour, requiring 30,000 lb. per hour of saturated steam at 40 lb. gage, 25,000 lb. per hour of which is returned to the power plant as condensate at 210 deg. This department will use 1,600 kw. of electrical energy and might be typical, but as to method of use only, of a paper mill dryer and beater load as a unit.

Product C: To be produced at the rate of 3 tons per hour, requiring 30,000 lb. per hour of saturated steam at 100 lb. gage, 25,000 lb. per hour of which is returned to the power plant as condensate at 210deg. This department will require 1,400 kw. of electrical energy and the product similar to product *B*.

Product D: To be produced at the rate of 2 tons per hour, absorbing 20,000 lb. per hr. of saturated steam at 100 lb. gage and using 500 kw. This product might be similar to product *A*.

The flow sheet below, indicates diagrammatically all of the departments *A, B, C* and *D* and the power plant equipment of a feasible scheme. Neglecting minor steam and water losses for simplicity, the feed-water make-up must equal the sum of the losses of the various departments, or 50,000 lb. per hour. This make-up feed water can be provided by several methods, but for supply water of certain character it may prove best to employ evaporators, especially if the arrangement as shown on the flow sheet is possible. In this scheme of operation, the first-effect coil is supplied with 100-lb.-gage process steam, while the last effect shell discharges the steam generated into the 40-lb.-gage process system, thus requiring no condenser.

The flow of steam and water in the evaporator plant for any method of evaporator operation, that is, double, triple or quadruple effect, requiring a lengthy detail, will be omitted for brevity. It will be noted that here, as in the plant discussed by the writer in the February 17, 1931, number of *Power*, entitled "The Paraffine Companies, Inc., Balance Power and Steam with Variable Effect Evaporators," if the method of evaporator operation, that is, number of effects in series, is changed, the amount of power generated by turbine-bled steam will be affected, thereby varying the remainder of electrical load that must be generated condensing. Naturally, the power plant operator should endeavor to generate as much of the electrical load demand as possible, operating the turbines non-condensing or bleeding rather than condensing, as the heat cost for the latter method is, roughly, four times the former.

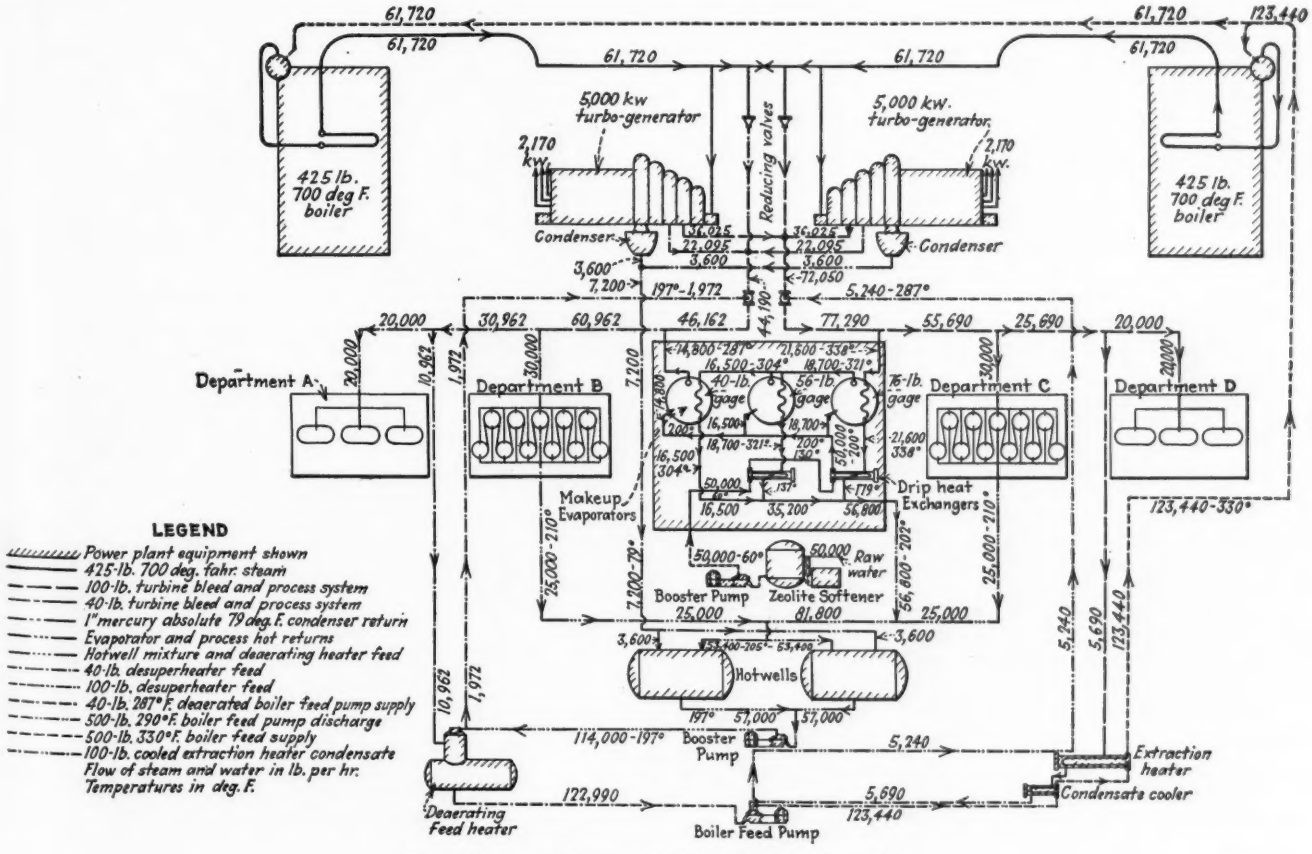
Continual reference to the flow sheet, below, is again stressed. Since the condition of the steam being bled from the turbine at 100 lb. and 40 lb. gage and also that exhausted into the condenser at 1 in. Hg.abs. is necessary to make other calculations, the detail necessary to obtain these quantities for corresponding pressures follow:

Steam at 425 lb. gage, 700 deg. total temperature, has a total heat of 1,360 B.t.u. per pound, is superheated 245 deg. and has an entropy of 1.63.

If the turbine could operate on the Rankine cycle, using the greatest possible amount of heat from each pound of steam, its cycle would be called adiabatic expansion at constant entropy, and the total heat of the steam bled at 100 lb. gage would be 1,222 B.t.u. per pound. This total heat subtracted from the initial total heat of 1,360 B.t.u. leaves a difference of 138 B.t.u. per pound, which is the amount that would be extracted from each pound of steam by an engine operating on the Rankine cycle.

Such an engine is ideal, the efficiency of which cannot be paralleled in actual practice. It is found from tests that an actual turbine of a certain size and design, when operating between the limits of 425 lb. 700 deg. and 100 lb. gage, is only about 75 per cent as efficient as the ideal Rankine engine, and as such would extract from each pound of steam 75 per cent $\times$ 138 = 103.5 B.t.u. per pound.

Subtracting this figure from the initial total heat of 1,360 B.t.u. results in the total heat of the steam bled from the turbine at 100 lb. gage, or 1,256.5 B.t.u. per pound at a total temperature of 461 deg. and a superheat of 123 deg. The total heat of 100-lb. saturated steam



Flow diagram of assumed industrial plant supplying power and steam for manufacturing purposes

is 1,189 B.t.u. per pound at a temperature of 338 degrees.

As the heat equivalent of a kilowatt-hour is 3,412 B.t.u., if the combined mechanical and generator efficiency of the turbine is 92 per cent, and since the turbine extracts from each pound of steam 103.5 B.t.u., it follows that to generate 1 kw.-hr. there must be bled from the turbine at 100 lb. gage $\frac{3,412}{103.5 \times .92}$ or 35.8 pounds.

It is important to note that with steam passing through a turbine at any pressure, generates a kilowatt-hour by extracting heat from the steam only in the amount of the heat equivalent of a kilowatt-hour divided by the combined mechanical and generator efficiency, neglecting radiation losses. Assuming this efficiency as 92 per cent, the heat required per kilowatt-hour generated will be $\frac{3,412,}{0.92}$ or 3,710 B.t.u.

Similarly, to determine the steam rate and state of steam bled from the turbine at 40 lb. gage the detail is as follows:

Total heat of 425 lb., 700 deg. steam is 1,360 B.t.u. per pound. Total heat of steam, if expanded through turbine at constant entropy to 40 lb. gage is 1,160 B.t.u. per pound. Possible heat extract by ideal engine is 200 B.t.u. per pound. Assuming a Rankine cycle ratio of 0.70, the actual turbine would extract 140 B.t.u. per pound, and if subtracted from the initial condition of 1,360 B.t.u. per pound, results in the 40 lb. bled steam having a heat content of 1,220 B.t.u. per pound and a total temperature of 373 deg. and a superheat of 85 deg. while the total heat of 40 lb. saturated steam is 1,175 B.t.u. per pound at a temperature of 287 deg. Similar to that for the 100 lb. bled steam analysis, the steam rate per kilowatt-hour for that steam bled at 40 lb. gage is $\frac{3,412}{140 \times 0.92}$ or 26.5 pounds.

For that steam expanded in the turbine to the extreme condition of one inch of mercury absolute, the detail would be: Total heat of 425 lb., 700 deg. steam, 1,360 B.t.u. per pound; total heat of the steam, if expanded at constant entropy to 1 in. Hg. abs., 876 B.t.u. per pound; possible heat extraction by an ideal Rankine engine, 484 B.t.u. per pound. Assuming a Rankine cycle ratio of 0.70, the actual turbine would extract 338.8 B.t.u. per pound. This heat extraction for generating electrical energy subtracted from the initial condition of 1,360 B.t.u. per pound results in the steam discharged to the condenser at 1 in. Hg. abs. having a heat content of 1,021.2 B.t.u. per pound at a temperature of 79 deg. and a moisture content of 7 per cent.

The steam required per kilowatt-hour generated by condensing operation will be $\frac{3,412}{338.8 \times .92,}$ or 10.95 lb. Since that steam used for turbine condensing operation is condensed and leaves the condenser as water at 79 deg., the condensate will have a heat content of 79 — 32, or 47 B.t.u. per pound. The heat change in each pound of steam used in condensing operation is 1,360 — 47, or 1,313 B.t.u. per pound, and the heat required per kilowatt-hour generated is 10.95 × 1,313, or 14,380 B.t.u.

Total electrical load to be carried by the power plant is as follows:

	Kw
Power plant auxiliaries	240
Dept. A requires	600
Dept. B requires	1,600
Dept. C requires	1,400
Dept. D requires	500
Total generated load	4,340

The sum of the amounts of power generated by bled steam plus that generated under condensing operation must equal the total electrical requirement. These two quantities affect each other and each must be of a definite amount to attain a perfect heat balance. First, it is to be noted that as the amount of power generated by bled steam decreases, the amount of power generated condensing must increase. Second, a counter-acting balance is brought about by the increasing quantity of turbine condenser condensate, resulting in a lower hotwell mixture temperature. As the hotwell temperature decreases, the quantity of 40 lb. bled steam necessary in the deaërating heater must increase. This will increase the quantity of 40-lb. bled steam and the power it generates.

By making several assumptions as to power generated condensing and through this "cut-and-try" process, it is found that the amount of power that must be generated under turbine condensing operation is 658 kw., leaving a non-condensing, or bled, load of 3,682 kw. To generate 658 kw. condensing will require an hourly steam quantity of 10.95 × 658 = 7,200 pound.

From the flow sheet, page 141, data on evaporator drip quantities and temperatures are obtained and the temperature of the mixture can be determined as follows:

	Lb.	B.t.u.
Third effect	16,500 (304–32) =	4,490,000
Second effect	18,700 (137–32) =	1,963,000
First effect	21,600 (179–32) =	3,175,000
Total	56,800	9,628,000

The resultant mixture will have a unit heat content of $\frac{9,628,000}{56,800} = 170$ B.t.u. per pound and a temperature of 170 + 32 = 202 degrees.

The resultant temperature of mixed hot returns from Departments B and C and evaporator condensate is determined as follows:

	Lb.	B.t.u.
Dept. B	25,000 (210–32) =	4,450,000
Dept. C	25,000 (210–32) =	4,450,000
Evaporator condensate	56,800 (202–32) =	9,628,000
Total	106,800	18,528,000

Total heat of each pound of mixture going to hotwells is $\frac{18,528,000}{106,800} = 173$ B.t.u. per pound at a temperature of 173 + 32 = 205 deg. The condenser return will have a total heat content of 7,200 (79 — 32) = 338,000 B.t.u. The mixture in the hotwell is then 114,000 lb. having a total heat content of 18,866,000 B.t.u. The resultant mixture in the hotwell will therefore have a unit heat content of $\frac{18,866,000}{114,000}$ or, 165 B.t.u. per pound at a temperature of (165 + 32) 197 degrees.

Having determined the quantity and temperature of the hotwell mixture, the next step is to find the amount of this water required for desuperheating the 40-lb. turbine-bled steam, and the amount of 40 lb. gage saturated steam required in the deaërating heater to give a resultant water mixture temperature of 287 deg. or that corresponding to the saturation temperature of steam at 40 pounds.

To determine these two unknown quantities, certain short cuts may be resorted to, but which are not as direct or as easily followed as the method given here.

Let X = the amount of 197 deg. hotwell water fed to the 40-lb.-gage desuperheater through the branch from the deaërating heater feed-water main.

Let Y = the amount of 40 lb. saturated steam required in the deaërating heater to give a resultant mixed water temperature of 287 deg. The total heat content of the water fed to the heater (114,000 — X) × (197-32) plus the total heat content of the 40-lb.-gage saturated steam fed to the heater, $Y \times 1,175$, must equal the total heat content of the resultant deaërating heater mixture at 287 deg., and (114,000 — X + Y) × (287-32) = (114,000 — X) × (197-32) + 1175 Y. Solving: $Y = \frac{90}{920}$ (114,000 — X).

By like analysis the total heat content of the superheated steam bled from the turbine at 40 lb. and discharged into the desuperheater, (20,000 + 30,000-14,800 + Y — X) × 1,220 plus the total heat of the 197 deg. water fed to the desuperheater, $X \times$ (197-32) must equal the total heat content of the saturated steam passing out of the desuperheater, and (20,000 + 30,000 — 14,800 + Y) × 1,175 = 20,000 + 30,000 — 14,800 + Y — X) × 1,220 + $X \times$ (197-32). Solving: $X = \frac{45}{1055}$ (35,200 + Y). But above $Y = \frac{90}{920}$ (114,000 — X) and substituting the value of X, $Y = \frac{90}{920} \times [114,000 - \frac{45}{1055} \times (35,200 + Y)]$. Solving for Y, the steam fed to the heater equals 10,962 lb. per hour. Substituting this value of Y in the equation $X = \frac{45}{1,055} \times (35,200 + Y)$, X, the weight of 197 deg. water fed to the 40 lb. desuperheater is found to be 1,972 lb. per hour.

Marking these hourly flow quantities on the flow sheet, page 140, the quantity

DISTRIBUTION OF POWER, HEAT AND COSTS

Department	Production Tons per Hour	Power: Kw.	Power: Million B.t.u. per Hour Equivalent Required from Boilers	Heat: For Process Only Million B.t.u. per Hour	Heat: Total Heat Million B.t.u. per Hour	Heat: Total Heat Million B.t.u. per Ton	Steam: Quantity Absorbed or Lost Lb. per Hour	Costs: Hourly for Total Heat Including Fixed Charges and Labor	Costs: Hourly for Steam Absorbed or Lost and Replaced as Make-up Water	Costs: Hourly Total	Costs: Total Cost per Ton
A	2.5	600	3.384	22.94	26.324	10.530	20,000	6.380	0.636	7.016	2,806
B	4.0	1,600	9.024	30.80	39.824	9.956	5,000	9.652	0.159	9.811	2,452
C	3.0	1,400	7.896	31.22	39.116	13.039	5,000	9.478	0.159	9.637	3.212
D	2.0	500	2.820	23.22	26.040	13.020	20,000	6.315	0.636	6.951	3.476
Total	11.5	4,100	23.124	108.18	131.304	.....	50,000	31.825	1.590	33.415	

of 287 deg. water going to the boiler-feed pumps is found to be 122,990 lb. per hour.

The flow sheet indicates the use of a condensate cooler to cool the 100 lb. condensed steam or water from the extraction heater so that it will not flash when admitted to the boiler-feed pump between two of the first stages. Some heat head will be saved and less hotwell flashing and consequent steam loss will occur than if the extraction heater condensate is allowed to pass at high temperature directly into the hotwells.

The method of determining the 100 lb. desuperheater feed water and extraction or bleed heater steam quantities is similar to that for the 40 lb. system in the foregoing. The extraction heater and condensate cooler may be considered as a single unit.

Assuming an 8 deg. terminal temperature difference between 100 lb. steam and water leaving the extraction heater, this water would be heated to 330 deg. and have a total heat of (330-32) or 298 B.t.u. pound. The 100 lb. turbine bleed steam enters the desuperheater with a total heat of 1,257 B.t.u. per pound. Water for desuperheating enters the desuperheater at 287 deg., with a total heat of (287-32) or 255 B.t.u. per pound. The saturated steam will leave the 100 lb. desuperheater with a total heat of 1,189 B.t.u. per pound.

If, as in the case of the 40-lb.-gage desuperheater steam system, X equals the 287 deg. water fed hourly to the desuperheater and Y the hourly quantity of saturated steam fed to the extraction heater, the following two equations may be written: (A) $(122.990 - X) \times (287 - 32) + (Y \times 1,189) = (122,990 - X + Y) \times 330 - 32$ and (B) $(71,600 + Y - X) \times 1,257 + X \times (287 - 32) = (71,600 + Y) \times 1,189$. Solving equation (B): $X = \frac{68}{1,002} \times (71,600 + Y)$.

Substituting this value of X in equation (A), Y, the hourly quantity of 100 lb. saturated steam fed to the extraction heater becomes 5,690 lb. per hour. Substituting this value of $Y = 5,690$ in either equations (A) or (B), X, the hourly quantity of 287 deg. water fed to the desuperheater is found to be 5,240 lb. per hour.

Having determined the above hourly quantities, the remainder of the figures may be computed and marked on the flow sheet. From the flow sheet showing 72,050 lb. of 100 lb. steam bled from the turbine hourly and the steam rate of 35.8 lb. per kw.-hr. found on page 142 it follows that the power generated with this steam is $\frac{72,050}{35.8}$ or 2,012. kw.-hr. Likewise, bleeding 44,190 lb. per hour from the turbine at 40 lb. at a steam rate, derived on page 142, of 26.5 lb. per kw.-hr., we can generate an additional $\frac{44,190,}{26.5}$ or 1,670 kw.-hr. Total non-condensing, or bleed load is 3,682 kilowatt-hours.

On page 142 it was shown that power could be generated condensing at a steam rate of 10.95 lb. per kilowatt-hour, the turbine condensing load being 658 kw., making the total generated load 4,340 kw. Power plant auxiliary load is 240 kw., and the process electrical load is 4,100 kilowatts.

It was shown that the heat extraction necessary to generate 1 kw.-hr. was 3,710 B.t.u., and since 3,682 kw. is generated by bled steam the hourly heat requirement for this power is 3,682 × 3,710 or 13,640,000 B.t.u. To generate 1 kw.-hr. by turbine condensing operation the heat requirement is 14,380 B.t.u., and to generate 658 kw. requires hourly 658 × 14,380 = 9,450,000 B.t.u. Total heat requirement for power generation is 23,090,000 B.t.u.

The average heat requirement per kilowatt-hour generated is $\frac{23,090,000}{4,340} =$ 5,320 B.t.u. and on the basis of power delivered from the power plant is $\frac{23,090,000}{4,100}$ = 5,640 B.t.u. per kilowatt-hours.

Cost Distribution

Since product A requires 20,000 lb. per hour of 40 lb. saturated steam, all of which is absorbed, and the loss made up with 60 deg. water, the hourly heat charge should be 20,000 × (1,175 — 60 + 32) = 22,940,000 B.t.u. Product B, however, takes 30,000 lb. per hour of 40 lb. saturated steam having an initial total heat content of 30,000 × 1,175 = 35,250,000 B.t.u. This department having a 5,000 lb. per hour loss returns 25,000 lb. of water per hour at 210 deg. having a total heat of 25,000 × (210 — 32) = 4,450,000 B.t.u., and the net heat used is 30,800,000 B.t.u.

Product C takes 30,000 lb. per hr. of 100 lb. saturated steam, having an inital total heat content of 30,000 × 1,189 = 35,670,000 B.t.u. This department also, having a 5,000 lb. per hour loss, returns 25,000 lb. per hour of water at 210 deg. having a heat content of 25,000 × (210 — 32) = 4,450,000 B.t.u. The net heat used is 31,220,000 B.t.u.

Product D requires 20,000 lb. per hour of 100 lb. saturated steam, and, like product A, all the steam is absorbed, the loss being made up with 60 deg. water, for which the hourly total heat charge should be 20,000 × (1,189 — 60 + 32) = 23,220,000 B.t.u. Total hourly process heat requirement is then 108,180,000 B.t.u.

The hourly heat requirement for power generation of 4,340 kw. was found to be 23,090,000 B.t.u., and the total hourly heat requirement is 131,270,000 B.t.u.

As a check on this total hourly heat demand, we find the boiler delivers 123,440 lb. per hour, which is raised from water at 330 deg. to steam at 425 lb., 700 deg., thereby adding hourly 123,440 × (1,360 — 330 + 32) = 131,300,000 B.t.u. which agrees with the foregoing.

A fairly close detailed estimate will show that such a power plant as herein contemplated will cost $600,000. Because different methods of accounting are resorted to by various managements, this study will consider the power plant as a necessary auxiliary of the industrial to fulfill certain duties in producing heat and electrical energy, it being the vendor of both at its actual cost and not sharing in the profits of the industrial.

Interest on the power plant investment is not included in the fixed charges here, but the return on the total plant investment should be considered when determining the selling price of the products.

It may be found that the annual charge for depreciation, taxes and insurance is 5 per cent of the initial cost, or.....	$30,000
Maintenance and supplies should not exceed 0.5 per cent of the initial cost, or $3,000	
Plus $0.0007 per kw.-hr. generated 26,600	29,600
Operating personnel.....	20,000
Total annual charge.....	$79,600
This annual combined fixed and operating charge is equivalent to an hourly charge of	9.09
Since the power plant hourly heat demand is 131,300,000 B.t.u., this combined charge may be considered on a million B.t.u. basis as $\frac{9.09}{131.3}$, or.....	$0.0693 per million B.t.u.
this being a convenient form to use in charging the various products.	
Based on fuel oil costing 90 cents per barrel of 6,200,000 B.t.u. and a boiler efficiency of 84 per cent, the unit cost of heat derived from the boilers will be $\frac{0.90}{\frac{0.84 \times 6,200,000}{1,000,000}}$, or.....	0.173 per million B.t.u.
Combined total unit fixed, labor and fuel charges.....	0.2423 per million B.t.u.

On the assumption that the raw

make-up water that is to be zeolite-treated before evaporation has a hardness of 10 gr. per gallon in terms of calcium carbonate, the equivalent weight of calcium carbonate to be replaced per 1,000 gal. is $\frac{10 \times 1{,}000}{7{,}000} = 1.43$ pound.

From the molecular weights of calcium carbonate (100) and sodium chloride (Salt 58.5) it may be found that the ratio of salt required to calcium carbonate replaced by weight is 1.17 to 1, or theoretically the salt required to neutralize the hardness in 1,000 gal. of 10-gr. water will be $1.17 \times 1.43 = 1.672$ lb. per 1,000 gallons.

Due to the inefficiency of the zeolite in salt absorption, the salt requirement should be increased to 2 lb. per 1,000 gal. If salt costs $16 per ton, the unit cost for softening will be $16 per million gallons. Assuming a water cost of 22½c. per 1,000 gal., or $225 per million gallons, the combined unit cost is $241 per million gallons. Assuming a 10 per cent additional cost due to loss in backwashing and boiler feed after treatment, the various departments should be charged for condensate not returned at $1.1 \times 241 = \$265$ per million gal., or $\frac{0.265}{8.33} =$ $0.0318 per 1,000 pounds.

From the data arrived at showing the amount of heat required and steam absorbed or lost in manufacturing products *A*, *B*, *C*, and *D*, together with the power required, the heat rate per kilowatt-hour of 5,640 B.t.u., the combined unit cost of $0.2423 per million B.t.u. for fixed charges, labor and fuel oil and the unit cost of makeup water at $0.0318 per 1,000 lb. the table may be set up with the first, second and seventh columns a base.

Consider product *A* as an example: Its power heat requirement for 600 kw. of electrical energy at a heat rate of 5,640 B.t.u. is 3,384,000 B.t.u. per hour and the process heat requirement is 22,940,000 B.t.u. per hour, making a total heat requirement of 26,324,000 B.t.u. per hour. If producing 2.5 tons per hour the unit heat rate is $\frac{26{,}324{,}000}{2.5} =$ 10,530,000 B.t.u. per ton.

The cost of product *A* for fixed, labor and fuel charges determined on a unit heat basis is 26.32×0.2423 or $6.380 per hour. For the steam absorbed, and which must be replaced as boiler feed make-up, product *A* would have an additional charge of $\frac{20{,}000 \times 0.0318}{1{,}000} =$ $0.636 per hour. Total hourly cost of product *A* for heat, electricity and water is $7.016 per hour and the unit cost is $\frac{7.016}{2.5} = \$2.806$ per ton.

Similarly, the data may be computed for products *B*, *C* and *D*, and the totals checked as follows:

The net heat added by the boilers being 131,300,000 B.t.u. per hour at a unit cost for fuel oil of only $0.173 per million B.t.u. costs $22.70 per hour. Fixed charges and labor are $9.09 per hour. The cost of make-up water $\frac{50{,}000 \times 0.0318}{1{,}000}$ is $1.59 per hour. Total hourly cost is $33.38 per hour. This checks with the total cost shown in the table as $33.415 per hour.

The particular power plant for any industrial should be chosen only after a thorough study of all the facts and conditions affecting the ultimate cost of the finished products to the degree dictated by funds available for fineness of performance.

▼ ▼ ▼

The Engineer's Responsibility In Smoke Abatement

AT A MEETING of the Metropolitan Section of the A.S.M.E., on January 19, Colonel Whitlock, research professor at Stevens Institute, discussed the question of smoke abatement, which he divided into three parts: elimination of sulphur fumes, fly ash and combustion equipment.

The progress made by the engineer in developing methods and equipment for eliminating sulphur fumes from stack gas has been meager, and much work is still needed in that direction. Some equipment has been developed in Europe, but it is expensive and not entirely practicable.

Greater progress has been made in methods and equipment for removal of fly ash and from flue gas. The most efficient equipment now available is the electrical precipitator. Cyclone collectors and cinder traps, according to Colonel Whitlock, have met with some success, but further improvement is necessary. The main difficulty with these types of collectors is that they depend upon the mass of the particles for separation and that the mass of the finer particles is too small to be separated out. Bag filters cannot generally be used, because the gas temperatures are too high.

The engineer's responsibility in this phase of smoke abatement is to further develop fly ash collecting equipment so that it will be available to the small plant.

Proper Installation Essential

Engineers have developed combustion equipment with which smokeless combustion can be obtained when properly installed and when the fuel is used for which it was designated. But when bituminous coal is burned in equipment designed to burn anthracite, or when improper installation is made, smoke results. Thus the responsibility of the engineer goes further than merely developing equipment for smokeless combustion; he should also see that proper installations are made. The alternative is city regulation.

The worst smoke offenders, Colonel Whitlock maintained, are plants of less than 200 horsepower.

To show experimentally the saving possible by smokeless combustion, slides were exhibited of a tea kettle containing a known weight of water and heated by a bunsen burner supplied with gas at a measured constant rate. In one case, with the burner placed so close to the kettle that soot deposits resulted, 10 minutes was required to heat the water, but when the burner was lowered so that soot was not deposited only eight minutes was required to heat the water to the same temperature with the same rate of gas flow to the burner, resulting in a saving of 20 per cent in fuel. This also illustrates what happens when the boiler-heating surface is too close to the fire.

The progress made in smoke abatement in Hudson County, N. J., was illustrated by the improvement made by the railroads at their New Jersey terminals. Prior to May of last year these terminals were the worst offenders, the maximum smoke reading for one railroad terminal being 38 units and the minimum being 10.6 units. In December, due to the efforts of a Railroad Smoke Association and the Smoke Commissioner, the maximum smoke reading had been reduced to 12.5, with a minimum of 2.4 units. During this time the average was reduced from 23.5 to 6.9 units.

H. B. Reynolds of the Interborough Rapid Transit Company, New York, in discussing Colonel Whitlock's paper, told of the work done at its 59th St. plant to reduce fly ash and smoke. In 1928 sixteen pounds of fly ash per 100 sq.ft. was collected per month from the roofs of near-by buildings, but after cinder catchers were installed, the latter part of 1928, this was reduced to one pound per 100 sq.ft. per month. A reduction in the amount of smoke was also obtained by providing overfire air and instructing the firemen.

Nuisance Poorly Defined

The inadequacy of the present New York City smoke ordinance, which states that black smoke shall not be emitted from stacks, was pointed out by Mr. Ricketts of the New York Edison Company. This definition of a smoke nuisance does not take into consideration the diameter of the smoke stream, the background, and whether the sun is shining on the smoke or through it, all of which factors influence the apparent blackness of the smoke. He also pointed out the necessity of a definition of the dust loading of flue gas in grains per cubic foot that would constitute a nuisance.

George Orrok remarked that it was possible to get just as pure an atmosphere as we are willing to pay for.

WHAT'S NEW IN PLANT EQUIPMENT

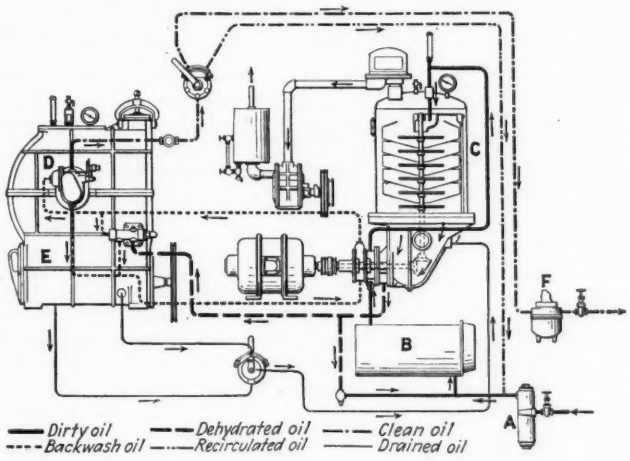
Unipass Electrical Oil Reclaimer Employing Dehydrator and Revolving Filter

DESIGNED especially for reclaiming electrical oils, the unipass unit now being put out by S. F. Bowser & Company, Inc., Fort Wayne, Ind., embodies a number of important features not employed heretofore in units of this type.

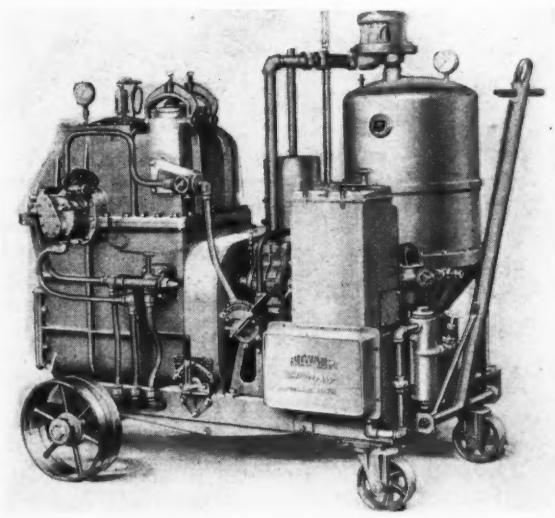
Referring to the schematic diagram, which shows the various parts of the unit with interconnections, incoming dirty oil enters a strainer *A* which removes rust and other foreign matter as well as the larger globules of water. From the strainer the oil passes through the electric heater *B* where the temperature is raised to approximately 120 deg. F. The heating elements are operated in conjunction with the motor starting switch and are individually controlled. From the heater the oil is pumped to the dehydrating chamber *C* maintained under a vacuum of 28 to 29 in. In this chamber a series of smooth-faced disks revolving at 1,200 r.p.m. disperse the oil in such a way that any entrained water is readily evaporated at the reduced boiling point. The watery vapor, together with other gases, is expelled directly to the atmosphere through a discharge on the vacuum pump.

After dehydration the oil collects in the sump of the chamber *C*, from which point it is pumped to an all-metallic filtering unit *D* designed to remove all suspended and sedimentary matter in a single pass. The filter element assembly comprises a series of eight twin noncorrosive porous metal plates embodying an infinite number of minute passages through which the filtration is accomplished. The element is rotated continuously at a speed of ½ r.p.m. Each twin plate is divided equally into six twin segments, and in operation one segment of each plate is automatically backwashed free of sludge while in the bottom position. A large-capacity sludge pan *E* is provided in the filter case for the accumulated sludge, carbon, etc., eliminating the necessity of frequent cleaning.

The filtered oil discharged by the five segments in the filtering position passes out to the filtered oil line through a positive-displacement meter *F* to storage or to the apparatus from which it was taken. All moving parts of the reclaimer, including pumps, vacuum pump agitation and dehydrator disks, are driven from a single motor. The entire electrical equipment is controlled by a hand-operated controller designed so that none of the electric heating units can be in operation while the pumps are idle.



Schematic diagram showing parts of unit with interconnections



In the Bowser unipass oil reclaimer all operating controls are on one side

To insure thorough cleansing when treating switch oils, which usually contain very finely divided carbon deposits, a pre-coat of porous filter-aid is introduced into the filtering compartment and applied to the surfaces of the metal plates. During this process filtration takes place through the entire filtering element, the backwashing feature being disengaged. Applying the filter-aid is said to be a simple operation requiring about three minutes.

Some of the claims made for this unit by the manufacturer are: Renders oil anhydrous and of high dielectric strength in a single pass; renders oil gas-free, including dissolved air and gases; reduces the organic acid content of serviced oil without introducing water or chemical solutions into the oil; and reduces the soluble and non-soluble sludges in the oil.

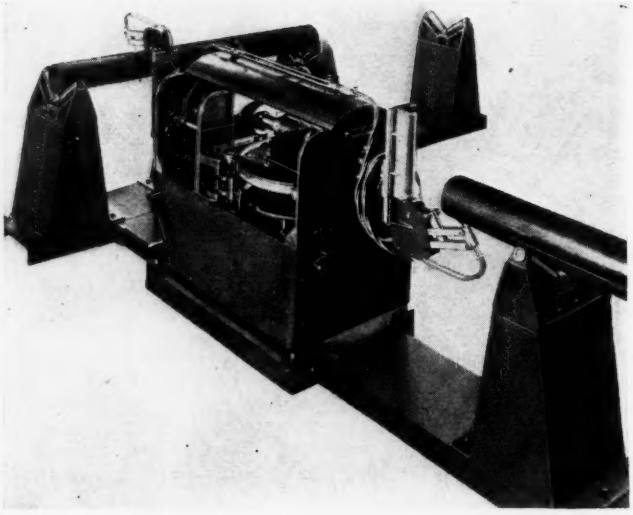
Douglass Pipe-Fitting Machine for Welded Connections

IN HIGH-PRESSURE and power piping installations, fabrication by welding has gained in favor so rapidly that the preparation of pipes accurately and uniformly for weld connections has become an operation of major importance. To facilitate and improve this work so that it may be done automatically without the use of templates or patterns, the new Douglass pipe-fitting machine shown in the illustration has been developed. This machine is being marketed by H. A. Brassert & Company, 310 South Michigan Ave., Chicago.

Clean cuts, correct bevels and uniformity of outline between the surfaces to be joined are the advantages emphasized for machine preparation. The new machine has been designed to cut automatically by oxy-acetylene torch pipe of 2 to 18 in. diameter for fabrication into welded tees, miter elbows and bull plugs. It also bevels plain end pipe for butt welds. The mechanical movement of the cutting torch is a combined rotating, longitudinal and angular motion, so that the correct bevel is produced in the pipe at any point. By the use of two heads on the machine the header pipe and the branch pipe may be cut simultaneously with the same generating mechanism, assuring a close fit of the two parts of the tee at the point of welding.

Bevels produced by the machine conform to the standards of the American Gas Association and the Heating and Piping Contractors' National Association. All cuts, it is claimed, are made smooth and clean, free from oxide and scale, so that parts as they come from the machine require no further preparation for welding.

Pipes can be cut on the machine in less time than it takes to lay out the



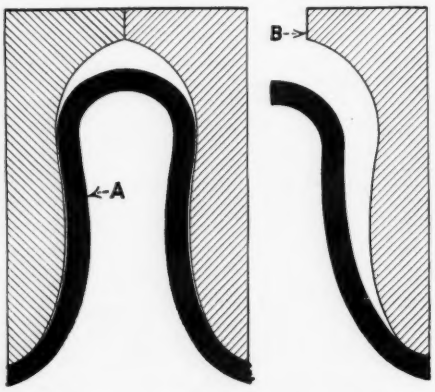
Douglass pipe-fitting machine simultaneously oxy-acetylene cutting a header and mating branch pipe

work for hand-cutting. No templates or patterns are needed, as the required curve produced by two intersecting cylinders of different diameters is generated in the machine itself. By two simple adjustments a complete range of pipe sizes is taken care of. Pipe standards or supports are of fixed height and the cutting machine itself is raised or lowered by four screw jacks actuated by a single crank. An indicator graduated in pipe diameters allows the correct setting to be made. Standard torch tips and torch valves are used. The latter are mounted on top of the machine, where they are easily accessible for adjustment.

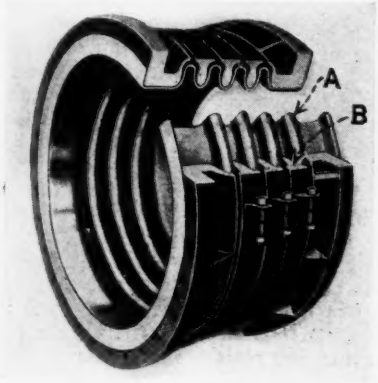
Improved Corrugated-Type Expansion Joint

DIRECTED FLEXING is an important feature of the improved corrugated expansion joint now being put out by E. B. Badger & Sons Company, Boston, Mass.

The improvements made to the joints consist, essentially, of matching the contour of equalizing rings *B* and the corrugations of flexing member *A* so that the latter assume the correct position at all pressures and temperatures, thus eliminating all localized flexing stresses and insuring uniform distribution of the flexing throughout the corrugations. The flexing member is now made of Everdur metal having a 96 per cent copper base, 3 per cent silicon and 1 per cent manganese. Several styles of joints are available, including the standard type with flanged ends, welding-end type, single unit, and welding-end type double unit with service outlet between the joints. Both the standard and the welding-end type can be equipped with monel sleeve for use with superheated steam.



View showing position taken by flexing element under expansion and contraction of pipe line

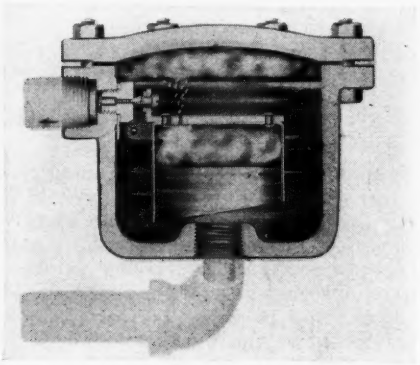


Cut-away view of Badger improved joint

Inverted Open Float Steam Trap

THE ILLUSTRATION shows an inverted open float steam trap being put out by the Crane Company, Chicago, Ill., in ½- and ¾-in. sizes for pressures up to 150 lb. The trap is intended for use with small steam-heated units such as heating coils, unit heaters, steam mains and risers and it can be used on both low-pressure and vacuum systems.

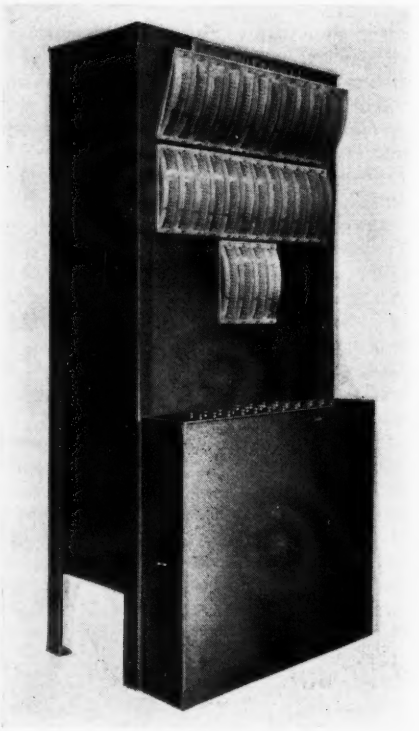
The float, which is fitted with a small vent hole for the release of air and other non-condensible gases, is drawn in one piece and has a needle-point valve mounted on the top as shown. With the inlet connected at the bottom center of the body the discharge can be turned to any desired angle. The trap is light in weight and can be supported by the connecting pipe lines. Capacities in pounds of condensation per hour for the ½- and ¾-in. sizes are, at 5 lb. pressure, 404 and 1,640, and for 150 lb. pressure 330 and 1,435 lb., respectively.



Crane inverted bucket trap shown in closed position

Panel for Stoker Zone Control

AS COMPANION equipment for zone-control underfeed stokers, the Hays Corporation of Michigan City, Ind., has developed the new control panel, with 28 specially designed differential-type draft gages, shown in the accompanying illustration. Design features are an airtight case and molded glass cover for each power unit, indicating pointers moving parallel to the scale faces to eliminate parallax, and operating mechanism of the standard colon leather type. Below the instruments may be seen the 28 zone damper buttons which give control of the primary air for combustion.



New Hays panel for zone control

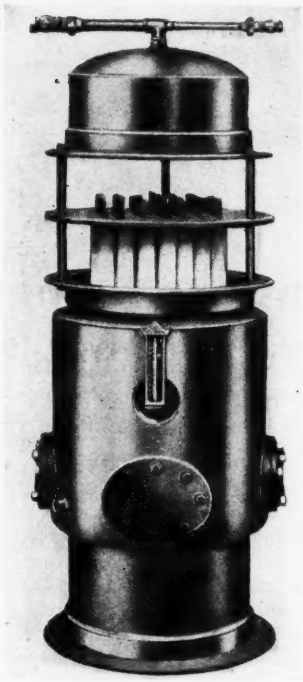


Fig. 1—Streamline filter unit partly open

Streamline Filter for Lubricating and Insulating Oil

THE STREAMLINE filter illustrated, now being put out by the Oliver United Filter Corporation, Inc., 33 West 42d St., New York City, is especially adapted to the removal of fine carbonaceous matter, water and other impurities from lubricating oils and the reconditioning of insulating oils.

The filter consists, essentially, of one or more columns of specially prepared paper disks mounted on a square rod or rods and compressed together under a spring tension of 80 lb. The paper disks are approximately 0.004 in. in thickness and the minute space between the disks together with the inequalities of the paper surfaces form extremely fine passages through which the oil to be filtered is forced under pressure up to 15 lb. or drawn by vacuum. The oil

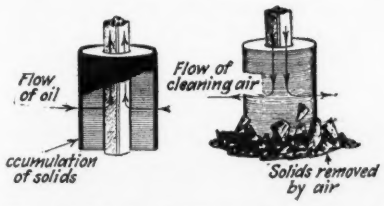


Fig. 2—View showing direction of flow of oil and reversed flow of compressed air for removing layer from outside surface

flow is from the outside of the disks to the spaces around the square rod. Thus the impurities in the oil are caused to form a layer or cake on the outside of the pack, Fig. 2, left.

The removal of this layer of impurities is readily done by passing compressed air at pressures up to 50 lb. through the filter in the opposite direction to normal flow. The flow of air through the filter packs tends to float off the layer of accumulated residue, which falls to the bottom, where it is ejected through a drain.

When installed on a diesel engine the used oil is passed through the filter either by a small gear pump driven off the engine shaft or bypassed from the oil-circulating pump on the engine. Provision for cleaning is made by a connection through a reducing nipple from the compressed air bottle employed for starting the unit. When necessary the filter can be heated by the exhaust gases of the engine, steam, hot water from engine jacket or electrically.

In the portable unit employed for the removal of fine colloidal carbon from and for the dehydration of insulating oils an electric immersion heater under thermostatic control is used.

Worm-Gear Reduction Unit

TO THE SERIES of standard worm-gear reduction units manufactured by the Cleveland Worm & Gear Company,

Cleveland worm-gear speed reducer

3258 East 80th St., Cleveland, Ohio, has been added the unit illustrated. Designed to transmit up to 7½ horsepower, the capacity depending on the reduction ratio and worm speed, the unit is available in both horizontal and vertical types. The horizontal units are built with the worm below or above the gear, and in the vertical units the gear shaft may extend upward or downward.

All shafts are in antifriction bearings, and the worm and its shaft are made from a single nickel-alloy steel forging. The worm thread is of the automotive type, with a high pressure angle. A range of nine standard ratios, varying from 4⅜ : 1 to 60 : 1 and a list of fifteen special ratios are available.

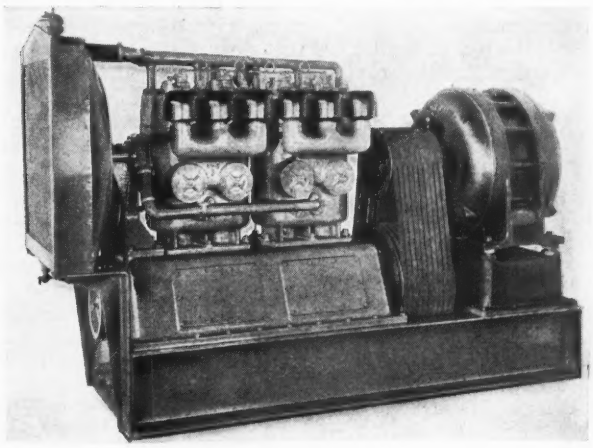
Supercharged Stationary Air Compressors

THE LINE of stationary air compressors incorporating the Rix patented supercharger principle, manufactured by the LeRoi Company, Milwaukee, Wis., is available in displacement capacities of 100, 200, 300, 400, 500, 600, 800, 1,000 and 1,200 cu.ft. per minute. With the supercharger principle an efficiency of 85 per cent is reached, it is claimed, at 100 lb. gage pressure.

The unit is mounted on a single base with its driving motor. A V-belt drive can be supplied or a direct-connected motor is available. If desired, to suit the floor space, the motor may be mounted alongside the air compressor, giving a horizontal, instead of a vertical, belt drive. In the unit illustrated an automotive radiator core is used for cooling, making a completely self-contained unit, but the compressor may be cooled with circulating water from the main if desired.

On its up stroke the trunk-type piston draws air into the annular space between the trunk and cylinder wall through an auxiliary suction valve. On the down stroke this air is trapped and compressed into the annular cored space in the cylinder wall between the water jackets. Just before the bottom of the stroke is reached the piston head uncovers a port and allows the compressed air to escape into the upper part of the cylinder, where it joins the air drawn into the cylinder in the normal way. In this way the volume of free air handled per stroke is increased 25 per cent and the volumetric efficiency raised proportionately.

Main bearings are bronzed-backed babbitt and connecting rod bearings centrifugally cast babbitt. Forced pressure lubrication through drilled holes in the crankshaft is supplied by a gear pump. Valves are of the plate disk type of alloy steel, hardened and ground.



Le Roi compressor with integral supercharger motor-driven through a V-belt

Cut-away section of 90-deg. Series 1R elbow

New Series Fittings for Pipe Welding

A NEW SERIES of fittings for pipe welding is announced by Tube-Turns, Inc., Louisville, Ky. Like the former series, the fittings included in the new line are made seamless with uniform wall thickness and radius. They are made with the same inside diameter and the same wall thickness and wall tolerance as the pipe with which they are used. This, together with the fact that the fittings are made of the same material as the pipe, is said to facilitate lining up and welding.

The complete line of fittings ("tube turns") now available includes 45- and 90-deg. elbows and 180-deg. return-type fittings for welding in sizes from $\frac{3}{4}$ in. to 20 in. and in practically all the weights required by modern piping.

Mechanical Boiler Water Level Indicator

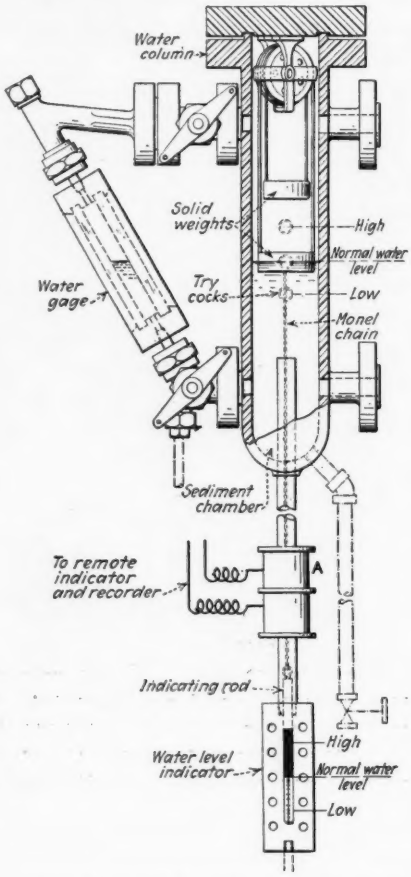
IN MODERN power plant boilers of large capacity the water gage must necessarily be a considerable distance above the operating floor. Although a water tender is generally stationed at the drum level, it is considered good practice to have, wherever possible, a reliable auxiliary water level gage or indicator so located that it can be observed readily by the boiler operator.

To meet the demand for a gage of this type the Yarnall-Waring Company, Chestnut Hill, Philadelphia, Pa., has introduced the "Eye-Line" gage illustrated. This gage is offered as a companion product to the Yarway Hi-Lo alarm water column and the Se-Sure inclined water gage, described in the Oct. 18, 1927, number of *Power*.

The gage employs displacement weights placed in the alarm water column as the actuating mechanism. A stainless-steel tube extends down from the column to the boiler room level, where it connects to a flat glass gage incorporating a movable plunger or target designed to indicate the water level in the boiler drum. This plunger is connected by a stainless-steel or monel-metal chain with the actuating balance weights in the water column, which is connected directly with the boiler drum.

As will be apparent from the illustration, the two counterweights are suspended from sheaves attached to the top of the column. The lower weight, from which the eye-level indicating plunger is suspended, rises and falls with the changes in water level in the column, thus causing a corresponding change in the position of the plunger in the eye-line gage.

In cases where it is desirable to have a remote indicator or recorder at some distance from the boiler an induction coil is mounted on the pipe above the indicator as shown at *A*. In this case an iron core is inserted inside a non-magnetic pipe and actuated by the chain supporting the eye-level indicating plunger. The induction coil is in turn

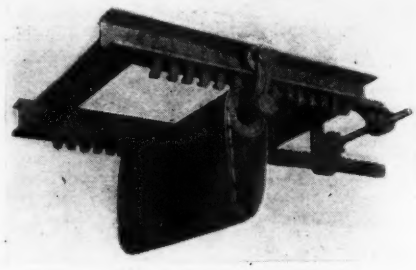


Yarway "Eye Line" water level indicator connected to Hi-Lo water alarm column

connected electrically with indicating or recording instruments located at a convenient point.

In locations where it may not be possible to drop the indicator vertically from the water column on the drum an auxiliary water column is used with horizontal extensions from the steam and water connections to bring the auxiliary water column to any desired position.

It is pointed out by the manufacturer that, as the water in the eye-line gage takes on and thereafter remains at practically room temperature, no erosion of the glass takes place and no mica protection is necessary. The absence of mica is said to give a clearer vision of the water level target inside the gage.



Apron gate in partly open position

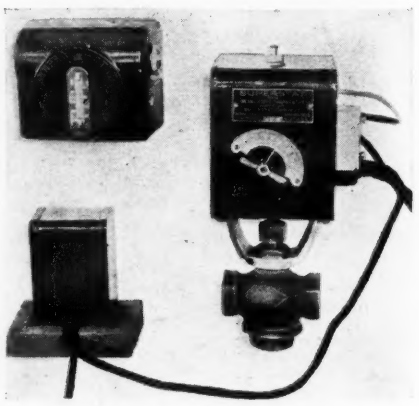
Bunker Apron-Type Gate

A COMPACT, light-weight, low-friction type apron gate has been introduced by the Stephens-Adamson Manufacturing Company, Aurora, Ill. A rack and pinion is used to open and close the gate, which it is claimed can easily be operated by one man. In opening, the carriage is pulled back, the lower rollers lift the cross bars one at a time and a cam disengages the hooks. In closing, the opposite action takes place: the bars are lifted into place by the rollers and the cam engages the hooks. The gate can be used with coal bunkers, and practically all classes of storage bins.

New Temperature Control

SUPERSTAT is the name given to a new temperature control introduced recently by the Superstat Company, Springfield, Mass. The control is available in two forms, a damper-type regulator, a unit having an operating arm which moves through an arc of 90 deg., and a valve modulator, an operating unit mounted directly on a globe valve for complete closure or on a butterfly-type valve for intermediate flow control. In both units the operative member has eleven operating positions within its maximum travel. The lever unit has a lifting power of 12 lb. on the arm, and the valve device has ample power to handle up to 4 in. valves on low pressures.

The control is powered by a reversible squirrel-cage motor and is designed to move in either direction from a given position upon $\frac{1}{8}$ of a deg. temperature change at the thermostatic element. Extreme sensitivity is one of the claims made for this control.



Valve operating unit with thermostat

NEWS of the FIELD

Creation of Federal Public Works Body Urged At Annual Meeting of Engineering Council

THE American Engineering Council at its annual meeting in Washington, D. C., Jan. 14 to 16, took steps to align the engineering profession in support of legislation to create an administration of public works, in accordance with the recommendation of President Hoover in his last message to Congress.

The council will work for the enactment by the present Congress of the bill introduced in the House by Representative Cochran of Missouri "to accelerate public construction in periods of business depression through the creation of an administration of public works and to provide for a more effective coordination and correlation of the public works functions of government."

The council, it was pointed out, has been advocating public works reorganization since it was founded in 1919 under the headship of Mr. Hoover. Favorable action by Congress is now believed to be probable owing to the disappointment of the people over the failure of public works activity to provide the measure of unemployment relief which had been expected.

Passage of this legislation, it was asserted in the council's discussion, will be a large factor in bringing about economies in the federal administration and in balancing the federal budget.

The Cochran bill provides for the appointment by the President of an administrator of public works to hold office for fifteen years at an annual salary of $15,000, and for the establishment in the administration of public works of a group of bureaus, each headed by a director, to administer the various public works services. The council will ask Congress to require that the administrator "shall by training and experience be qualified to administer the affairs of the administration of public works and to evaluate the technical principles and operations involved in the work."

The engineers will seek an amendment to the Cochran bill placing the design, construction, maintenance, operation and repair of ferries, barges, hospitals, lighthouses, prisons, storehouses, docks, and flood-control works under the control of the administrator of public works.

Another engineering amendment would enable the administrator to employ from time to time, by contract, and at the established rates of compensation, outside professional or technical service of competent persons, firms, or corporations, for the architectural and engineering designing, planning and construction of such federal buildings and works as are now or may, in the future, be placed under his jurisdiction.

William S. Lee

William S. Lee of Charlotte, N. C., past president of the American Institute of Electrical Engineers, was elected president of the council for two years, succeeding, Carl E. Grunsky of San Francisco. L. B. Stillwell and Gen. Richard C. Marshall, Jr., of New York were chosen vice-presidents. Farley Osgood of New York was named treasurer to succeed Dr. Harrison E. Howe of Washington.

New Boiler and License Law for Houston, Texas

AN ORDINANCE covering the inspection of boilers and elevators, and the issuance of engineers' licenses went into effect recently at Houston, Texas.

The new law creates the office of steam boiler and elevator inspector, consisting of an inspector and one or more assistants appointed by the Mayor with the consent of the City Council, and a Board of Examiners of Steam Engineers, consisting of three members similarly appointed.

Provision is made for the inspection of all boilers once a year and before installation, with the exception of those carrying a pressure of 15 lb. or less, used exclusively for heating private dwellings. Power-operated passenger and freight elevators are to be inspected every three months and hand-operated elevators every six months. All proposed changes affecting the safety of operation in either boiler or elevator installations must be approved by the inspector before being made. In all installations the ordinance recommends conformity with the rules and regulations of the A.S.M.E. Boiler Codes and Safety Code for Elevators.

The Board of Examiners will act as a court of appeals from all decisions of the inspector, and will conduct examinations for engineers' licenses. Four classes of licenses will be issued: First grade, to have charge of steam plants of any capacity; second grade, not exceeding 150 hp.; third grade, not over 40 hp.; and hoisting and portable engineers' license.

House Orders Investigation Of Public Utility Control

INVESTIGATION by the House Committee on Interstate Commerce into the ownership and control of public utilities was authorized by the House Jan. 19, when it adopted the resolution presented by Representative Rayburn, chairman of the committee. The inquiry will be conducted by Walter M. W. Splawn, committee counsel, and will get under way as soon as funds are provided. Chairman Rayburn is asking for a maximum of $100,000, pointing out that a similar inquiry last session into the ownership and control of railroads cost $39,755 out of an authorized expenditure of $50,000. It is anticipated that more expense will be incurred in the public utility investigation because the ramifications of ownership and control are believed to be much broader in that field than in railroads.

The inquiry follows the recommendation made by Mr. Splawn last session that before railroad holding companies be subjected to regulation, investigation should be made to determine whether public utility holding companies should not also be subjected to regulation. As explained by Chairman Rayburn the purpose of the inquiry is to develop the facts with respect to the control of power, pipe line and communication utilities to determine whether they should be placed under the Interstate Commerce Commission. No consideration will be given to the rates charged to the public for service.

Ohio Sues Tenn. Boiler Co. For Workman's Compensation

IN AN ATTEMPT to force an out-of-state corporation engaged in work within its borders to pay an award made under its workmen's compensation act, the State of Ohio has filed an original suit in the United States Supreme Court. It is provided in the Constitution that the Supreme Court shall have original jurisdiction over a controversy between a state and a citizen of another state. The case arose over an accident during the erection and installation of tanks at Ironton by the Chattanooga Boiler & Tank Company, which is a Tennessee corporation.

It is contended by Ohio that the boiler company neglected to comply with the workmen's compensation law of the state by becoming a subscriber to the state insurance fund. Following an accident involving the death of one of the workmen, the state awarded a compensation payment to the heirs of the deceased. This award was paid out of the state fund. The purpose of the suit is to recover the amount of the compensation, $4,910.64, from the company. Under the rules of the Supreme Court, the company will be allowed twenty days from the filing of bill of complaint to file its answering statement. The bill of complaint was entered on January 11.

Reclamation Bureau Favors Columbia Basin Development

FEASIBILITY of the proposed Columbia Basin power plant and irrigation project in the State of Washington, estimated to represent an ultimate total investment of $394,155,000, is upheld in a report of the Bureau of Reclamation engineers which Commissioner Elwood Mead made public Jan. 16.

The proposed project now before Congress includes a dam and power plant to cost $185,890,000, including interest during construction, which is to be paid from power revenues with 4 per cent interest within 50 years after completion.

The plans, it was asserted in Dr. Mead's letter, summarizing the conclusion of the report, contemplate the development of 2,100,000 hp., of which 1,100,000 hp. will be firm power. Irrigation costs will be defrayed in part by surplus power revenues and as construction charges by the land owners. The estimated selling price of firm power is 2.25 mills per kilowatt-hour at the switchboard. Secondary power is to be sold at a price that will result in a pumping charge of $1 per acre.

Disapproval of the project has been voiced by Board of Engineers for Rivers and Harbors of the War Department, on the grounds that there is no market for the large amount of power at present and that many years would be required for the development of such a market. Furthermore, the board believes that further cultivation of land is inadvisable since agriculture has suffered from over-production for some time.

Dr. Mead points out that the project could not be completed until 1940, at which time, he believes, the capacity of present power facilities in the Northwest will have been entirely absorbed. "Much will depend," he says, "on the attitude of the states and municipalities toward the enterprise."

COMING CONVENTIONS

American Society of Mechanical Engineers. Spring meeting at Bigwin Inn, Lake of Bays, Ont., Canada, June 27-July 1. Oil and Gas Power meeting at Pennsylvania State College, June 8-11. Secretary, Calvin W. Rice, 33 West 39th St., New York City.

American Institute of Electrical Engineers. District meetings at Milwaukee, Wis., March 14-16, and Providence, R. I., May 4-7. Annual summer convention at Cleveland, Ohio, June 20-24. Secretary, F. L. Hutchinson, 33 West 39th St., New York.

American Boiler Manufacturers Association. Mid-winter meeting at the Hotel Cleveland, Cleveland, Ohio, Feb. 11. Annual convention at the Greenbrier Hotel, White Sulphur Springs, W. Va., May 23-25. Secretary, A. C. Baker, 801 Rockefeller Building, Cleveland, Ohio.

American Society for Testing Materials. Annual meeting at Chalfonte-Haddon Hall, Atlantic City, N. J., June 20-24. Secretary, C. L. Warwick, 1315 Spruce St., Philadelphia, Pa.

American Water Works Association. Annual convention at the Hotel Peabody, Memphis, Tenn., May 2-6. Secretary, Beekman C. Little, 29 West 39th St., New York City.

Engineering Institute of Canada. Annual meeting at the Royal York Hotel, Toronto, Ont., Feb. 3-5. Secretary, R. J. Durley, 2050 Mansfield St., Montreal, Que.

National District Heating Association. Annual convention at the William Penn Hotel, Pittsburgh, Pa., June 14-17. Secretary, D. L. Gaskill, Greenville, Ohio.

National Marine Engineers Beneficial Association. Annual convention at the Hotel Jefferson, St. Louis, Mo., Feb. 8-12. Secretary-Treasurer, Albert L. Jones, 313 Machinists' Building, Washington, D. C.

National Electric Light Association. Annual convention in the Atlantic City Auditorium and Convention Hall, Atlantic City, N. J., June 6-10. Secretary, A. Jackson Marshall, 420 Lexington Ave., New York City.

SECTION MEETINGS

A.S.M.E., Metroplitan Section. Meeting in Room 502, Engineering Socities Building, Feb. 1 at 8 p.m. Subject: "High-Capacity Coal Pulverizers at Kips Bay Station," Perry Cassidy.

A.S.M.E., Schenectady Section. Meeting in Edison Club Hall, Feb. 4 at 8 p.m. Subject: "Air Conditioning in Buildings," A. R. Stevenson, Jr.

A.I.&S.E.E., Philadelphia Section. Meeting in the Engineers' Club, Feb. 6 at 7 p.m. Subject: "Mercury-Arc Rectifiers for Steel-Mill and Industrial Applications," W. E. Gutzwiller and R. W. Davis.

N.A.P.R.E. First Pacific Coast conference at the Civic Auditorium, San Francisco, Calif., Feb. 17-18. Washington, Oregon, California and Arizona chapters will take part.

Roosevelt Opposes Granting More Water to Niagara Co.

IN A LETTER presented to the Federal Power Commission at the opening hearing Jan. 18, Governor Franklin D. Roosevelt, of New York, expressed opposition to granting the application of the Niagara Falls Power Company for the diversion of 275 sec.ft. more water at the falls and the addition of $1,500,000 to its fixed capital.

By its present license which runs until 1971, the company has the right to draw 19,725 sec.ft. of water from the Niagara River. Recently it paid $1,500,000 to the Hydraulic Race Company and group of mill owners in Lockport, N. Y., for their rights to the additional 275 sec.ft., which it is now using under a temporary permit.

If the Power Commission grants the application, the company will have a monopoly of Niagara River water, as the additional 275 sec.ft. will make up the entire amount of 20,000 sec.ft. that the United States has the right to draw under the treaty of 1910 with Canada.

Governor Roosevelt in his letter contends that the licenses or permits held by the Hydraulic Race Company and the mill owners were not transferable. He also urged the commission, if it should grant the application, not to permit the power company to add $1,500,000 to its capital but to impose such conditions as will protect the public.

Northern States Franchise Refused by St. Paul Voters

BY A MAJORITY of approximately five to one, St. Paul voters defeated the three proposed franchises for the Northern States Power Company, submitted at a special election Jan. 12.

The proposed franchises would have given the company the right to use St. Paul streets for twenty years to furnish steam, gas and electric service in the city; and provided for the elimination of a 5 per cent gross earnings license fee which the city collected for the past 25 years under the old franchises, and also provided that St. Paul rates should be on a parity with Minneapolis rates for similar services.

The most strenuous objection on the part of the voters was to the elimination of the gross earnings tax, which in 1931, netted the city $383,000, and the franchises failed to carry in a single precinct. Throughout the campaign the mayor and other officials of the city emphasized their belief that the franchise should contain provisions for immediate valuation and appraisal of the company's properties in St. Paul as a basis for rate making.

Simultaneous with the defeat of the franchises in St. Paul, the voters of Marshall, Minn., at a special election Jan. 12, rejected the proposal for the sale of the municipal electric and steam

plant to the Northern States company by a majority of nearly 13 to 1. A franchise to operate in the city for twenty years would have been given the power company, had its proposal been accepted.

E.I.C. Annual Meeting at Toronto, Ont., Feb. 3-5

ENGINEERING INSTITUTE OF CANADA will hold its 46th annual general meeting this year in Toronto, Ont., from Feb. 3 to 5 inclusive. Headquarters for the meeting will be established at the Royal York Hotel, where business and technical sessions, entertainment features and social events will be held. In addition, several trips to engineering works in the vicinity are scheduled.

At the technical sessions a wide range of topics will be discussed, among which are: "Hydro-Electric Developments on the Lievre River"; "Ice Pressure on Engineering Structures"; "The Utilization of Low-Grade Fuels"; and "Central Heating at the Toronto Terminal and at Winnipeg."

Herndon Steam Plant Suit Over Water Use Dismissed

DISMISSAL of a suit through the filing of a stipulation in the Superior Court in Fresno, Calif., has removed one of the obstacles in the way of construction of the San Joaquin Light & Power Corporation's proposed 50,000-kw. steam plant at Herndon, ten miles north of Fresno. The suit, which was brought about a year ago by the San Joaquin & Kings River Canal & Irrigation Company against the Lisenby Estate Company, from which the power company had purchased the property, involved riparian rights on the power plant site and sought to prevent the power company from taking water from the San Joaquin River for use in the steam plant. The stipulation contained the provision that the power company may use the water from the river as long as such use does not materially interfere with the flow of the river.

No definite date has been set by company officials for the start of construction of the Herndon steam plant, although early spring has been tentatively mentioned, with the object of having the plant ready for operation by the summer of 1933.

How's Business?

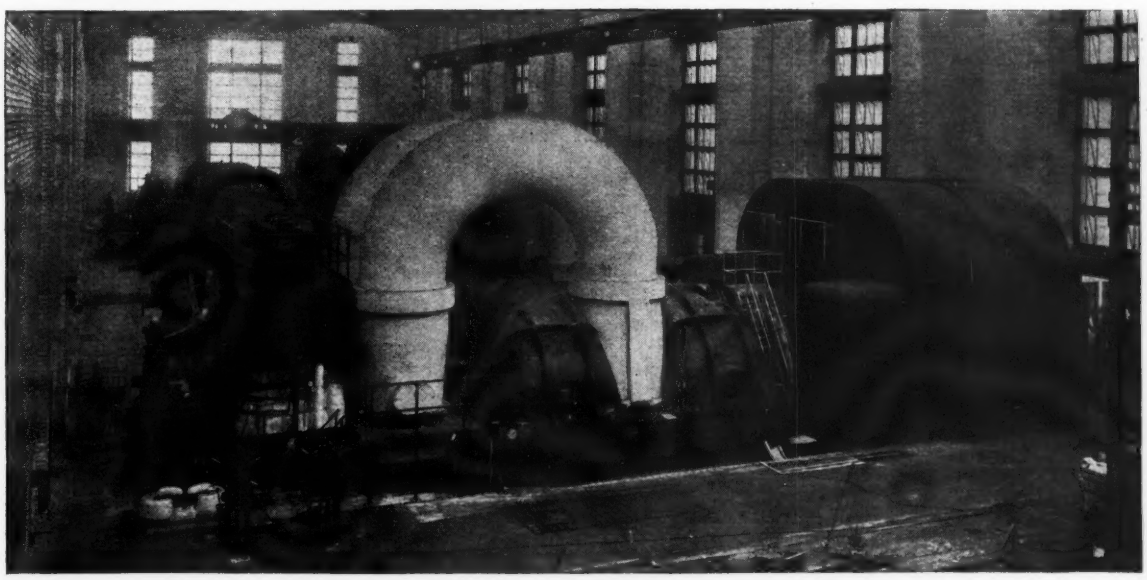
SIGNS of seasonal acceleration in activity are still slight, excepting only in steel, which is being mildly stimulated by slowly increasing automobile output and scattered demand from other sources. Although our index has risen for two successive weeks, merchandise movement by rail is unseasonably slack, and declining electric power production (1,602,482,000 kw.-hr. for the week ended Jan. 16, according to the N.E.L.A.) is a disappointing indication of continued industrial stagnation and diminishing domestic consumption. The widening wave of wage cuts and retrenchment in public expenditure projects is steadily contracting consumer purchasing power in a vicious circle. Except for the striking strength in wheat and cotton, commodity prices are still saggy.—*The Business Week,* Jan. 27.

Obituary

Word has just been received that JAMES MARKHAM, for many years a prominent member of the Providence Association of the National Association of Power Engineers, died on Dec. 14 at his home in Belmont, Mass., after an illness of fifteen months.

WILLIAM D. POMEROY, general manager, vice-president and director of Goulds Pumps, Inc., Seneca Falls, N. Y., died on Jan. 6 at the age of 57 years. He had been ill for more than a year, suffering from a nervous breakdown and severe heart attacks. Born in Utica, N. Y., Mr. Pomeroy was graduated from Cornell University in 1895. From 1901 to 1907 he was employed by the Bullock Electric Company, a subsidiary of Allis-Chalmers Manufacturing Company, where he advanced from draftsman to works manager. In 1907 he joined the Goulds company as works manager and was made general manager in 1915. A member of the A.S.M.E. and A.I.E.E., Mr. Pomeroy was widely known in manufacturing and engineering circles.

FIRST 160,000-KW. UNIT IN SERVICE AT HUDSON AVENUE



Recently placed in operation, this new unit is the first of two 160,000-kw. tandem-compound turbine-generators for the Hudson Avenue station of the Brooklyn Edison Company. It is supplied with steam at 400 lb. pressure and 730 deg. F. temperature by four boilers, each with a capacity of 530,000 lb. per hour, fired by the longest under-feed stokers yet made. The turbine exhausts into the largest single-pass condenser, which has a 101,000-sq.ft. surface. Rated at 200,000 kva., the generator has the biggest 1,800-r.p.m. revolving field ever built. Completion of both units will raise the total installation of the station to 770,000 kw., giving it the greatest capacity of any steam plant in the world

NEW PLANT CONSTRUCTION

COMPILED BY THE MCGRAW-HILL BUSINESS NEWS DEPARTMENT, WHICH IS PREPARED TO FURNISH A MORE COMPLETE DAILY SERVICE TO THOSE WHO WISH IT

Ala., Mobile—Treasury Department, Office of Supervising Architect, will receive bids until January 29 for addition to Marine Hospital including nurses' quarters, central steam heating plant, boiler house, etc., here. Estimated cost $415,000. Warren, Knight & Davis, Protective Life Bldg., Birmingham, and Chandler P. Younge, Merchants National Bank Bldg., Mobile, are associate architects.

Arkansas—Mississippi Valley Power Co., will start activities in near future on construction program. Major projects include rebuilding three miles of 33,000 va. line running out of Fort Smith and installation of additional regulators on two principal sub-stations. Distribution system will be rebuilt or partially rebuilt in Fort Smith, South Fort Smith, Van Buren, Charleston, Ratcliff, Alma, Dyar, Lamar and Ozark.

Ill., Collinsville — City Council rejected proposition for waterworks improvements consisting of well, pump house, electrically driven deep well pump, mains, etc. $168,000. Kinsey Engineering Co., Marshall Bank Bldg., Pekin, Engr.

Ill., Peoria Heights (br. Peoria)—City, c/o Paul R. Meyer, Clk., plans construction of waterworks system including well, pump house, pumps, tank on tower, etc. Estimated cost $100,000. W. A. Fuller Co., 2916 Shenandoah Ave., St. Louis, Mo., is engineer. Owner will soon receive bids on test wells.

Kan., Oswego—City voted $35,000 bonds for construction of municipal gas plant. Engineer not selected.

Ky., Mount Sterling — City postponed construction of electric light and power plant. Burns & McDonnell Engineering Co., 400 Interstate Bldg., Kansas City, Mo., Engr. Project temporarily abandoned.

La., Shreveport — G. E. Lamb, Constructing Quartermaster, War Department, will soon receive bids for construction of a sewage treatment plant, also installation of underground conduit system, etc., at Barkdale Field.

Md., Baltimore. — Palmer & Lamdin, 613 North Charles St., Archts., will receive bids until February 2 for construction of a 10 story apartment building including steam heating system, etc., at Charles St. and Bishops' Road, for Northway Apartment Co., I. C. Goldstein, Pres., Calvert Bldg. Estimated cost $850,000.

Mass., Provincetown—Northern Gas Utilities Co., 15 Congress St., Boston, revising plans for construction of a gas plant. Estimated cost $70,000. George H. Stevens, 141 Milk St., Boston, is architect.

Minn., St. Paul—U. S. Engineer's Office, War Department, 615 Commerce Bldg., awarded contract for construction of a power house at Twin City lock and dam to E. J. Dunnigan, 1014 Builders Exchange.

Mo., St. Louis—Board of Public Service, will receive bids about March 15 for construction of first unit of negro city hospital to consist of service building, boiler house, laundry building, etc., at Goode Ave., St. Ferdinand, Kennerly and Whittier Sts. Estimated cost $350,000. A. Osburg, 301 City Hall, is architect. L. R. Bowen, 301 City Hall, is engineer.

Mo., Sullivan—City, N. M. Shaffer, Clk., will soon receive new bids for construction of 43 x 150 ft. power house, 15,000 gal. oil storage tank and water cooling tower for circulation water in connection light and power plant. W. A. Fuller Co., 2916 Shenandoah Ave., St. Louis, are engineers. Former bids rejected. Equipment contracts awarded.

N. J., Scotch Plains—Board of Chosen Freeholders, Court House, Elizabeth, awarded general contract for construction of a 1 story addition to power house and garage building at Bonnie Burns Sanitorium to Robert McLeod, Inc., 255 Main St., East Orange. Estimated cost $40,000.

O., Cleveland—John Carroll University, H. R. Neff, Chn. of Bldg. Committee, Marshall Bldg., plans construction of a group of buildings, including physics and biology building, chemistry building, students dormitory, etc., at Miramer Blvd. Estimated cost $4,000,000. Philip L. Small, Inc., Terminal Tower, are architects. George S. Rider Co., Marshall Bldg., are engineers.

O., Xenia—Ohio Soldiers & Sailors Orphans Home, will soon award contract for construction of refrigeration plant. Contract awarded for dining hall, kitchen and storeroom to Galbreath & Leonard, 42 East Gay St., Columbus, $85,000.

Ore., Klamath Falls — California Oregon Power Co., 58 Sutter St., San Francisco, Calif., will build a 1,500 kw. sub-station, also twenty-two mile transmission line here by owner's forces. Estimated cost $40,000.

Pa., Rostraver — United States Engineer, Keenan Bldg., Pittsburgh, will receive bids about February 1 for construction of two power houses and one administration building at Lock No. 4 here. Estimated cost $34,000. Private plans.

Pa., Wernersville—Department of Property & Supplies, John L. Hanna, Secy., Harrisburg, will soon award contract for construction of a refrigeration plant including compressor, motor, pumps, etc., at Wernersville State Hospital here. W. H. Derchant & Sons, 632 Washington St., Reading, are engineers.

Washington—U. S. Reclamation Bureau, R. F. Walter, Ch. Engr., reports irrigation and power project for the Columbia River Basin, 150 mi. below Canadian border has been mapped out and specifications compiled and report before Department of Interior, Washington. Plans include dam 4,100 ft. long involving more than 11,000,000 cu.yd. of concrete 100 miles northwest of Spokane, will raise height of river 350 ft. and will back up a reservoir 150 miles from Grand Coulee to Canadian border to irrigate 1,200,000 acres.

Equipment Wanted

Pump—Mare Island (sta. Vallejo), **Calif.**—Bureau of Supplies & Accounts, Navy Department, Washington, D. C., will receive bids until February 2 for two electrically driven fire pumps and one electric flushing pump for Navy Yard here. Sch. 7268.

Motor, Switch, Boiler, etc.—Washington, D. C.—General Purchasing Officer of Panama Canal, will receive bids until February 3 for electric motors, motor starting switches, two circuit panels, boilers, boiler tubes, etc. Sch. 2720.

Pumps—Versailles, Ky.—City will receive bids about April 1 for two 250 g.p.m. centrifugal pumps, with 60 hp. motors, automatic control in connection with auxiliary raw water supply on Kentucky River. Estimated cost $50,000.

Coal Bunker, Stoker Equipment, etc.—Bedford, Mass.—Veterans' Bureau, c/o Col. L. H. Tripp, Arlington Bldg., Washington, D. C., will receive bids until February 9 for coal bunker, stoker equipment, etc., for Veterans' Administration Hospital here.

Motor—Cambridge (Boston P. O.), **Mass.**—J. L. Gleason & Co., 207 Bent St., plans to purchase a 25 hp. motor, 550 v., 3 ph., 900 or 1,200 r.p.m.

Pumps—Marquette, Mich.—City County, J. F. Anderson, Purch. Agt., will receive bids until February 23 for installation of three new electric pumps, also removal of existing boilers, pumps, etc., in connection with waterworks improvements.

Pumps — Puget Sound, Wash. — Bureau of Supplies & Accounts, Navy Department, Washington, D. C., will receive bids until February 2 for four main motor air-cooled circulation pumps and spares. Sch. 7202.

Engines—Huntington, W. Va.—U. S. Engineer Office, War Department, will receive bids until January 29 for five steam hoisting engines without boiler, single drum, non-reversing for Locks Nos. 14, 15, 16, 17, 28 on Ohio River.

Pumps, Turbines, Motors and Electrical Equipment—Calgary, Alta.—City Commissioners, c/o A. G. Graves, City Hall, will receive bids until February 12 for furnishing and installing centrifugal pumps with hydraulic turbines, motors and electrical equipment, etc., for Glenmore Pump and Screen House.

Electrical Equipment—Toronto, Ont.—Arrow-Hart & Hegeman Electric Co., Inc., 103 Hawthorne St., Hartford, Conn., plans to purchase electrical equipment for recently acquired building on Spadina Ave. here.

Pumping Unit—Wellington, N. Z.—City Clerk will receive bids until February 11 for supply of one electrically driven centrifugal pumping unit with all fittings, accessories and spares.

Transformers — Wellington, N. Z. — Public Works, Supplies and Tenders Committee, will receive bids until February 23 for four 1,000 kva. transformers complete with automatic tap changing equipment, also until May 3 for supply of 110 kva. and 33 kva. outside switch gear and steel work.

Industrial Projects

Calif., Los Angeles—Chrysler Corp., Walter P. Chrysler, 341 Massachusetts Ave., Detroit, Mich., have selected a 30 acre site at Eastland and Slauson Aves. here for automobile assembly plant, 300,000 sq.ft. Plans call for development of a parts stock department with expenditures for tools, equipment, assembly lines and other machinery. Estimated cost $2,000,000 to $3,000,000. Architect not selected. Chrysler, Plymouth, DeSoto and Dodge passenger cars and Graham trucks will be assembled for Pacific Coast in main plant.

Calif., Los Angeles—Willard Storage Battery Co., 5700 East 9th St., will receive bids about January 30, for addition to battery plant including erection of lead oxide mill, acid-reclaiming plant and enlargement of administration building. Estimated cost $100,000.

Conn., Manchester—L. T. Wood Co., Bissell St., awarded contract for construction of an ice manufacturing plant to G. Schreiber & Son, Inc., 285 West Centre St., South Manchester. Estimated cost $75,000.

Conn., Meriden—International Silver Co., J. A. Hutchinson, 48 State St., plans construction of a 1 story, 45 x 300 ft. plant. Estimated cost to exceed $40,000. Private plans. Maturity indefinite.

Ia., Dubuque — Maizewood Building Corp., awarded contract for rebuilding 65 x 225 ft. factory and 65 x 129 ft. warehouse destroyed by fire, to Charles Yokkum, Dubuque. $40,000. New equipment to cost $75,000 will be installed.

Louisiana and Texas—Continental Can Co., 1 Pershing Sq., New York, N. Y., will soon award contract for construction of a can manufacturing plant at Harvey, La. Estimated cost $125,000. Favrot & Livaudais, Nola Bldg., Harvey, Archts. Also plans plant at Houston, Tex. Exceeds $40,000.

Mich., Ypsilanti—Ford Motor Co., Construction Department, Dearborn, awarded contract for a 1 story, 160 x 309 ft. factory building as unit to automobile parts factory to H. G. Christman Burke Co., 1010 Fisher Bldg., Detroit. Estimated cost $150,000. Conveying system, etc., will be installed.

N. J., East Rutherford—Flintkote Co., Maple St., will soon award contract for construction of a 2 story factory. Estimated cost $40,000. Karl A. Lefren, Inc., 171 Madison Ave., New York, N. Y., are architects and engineers.

N. J., Lakewood — Lakewood Comet Aero Advertising Co., H. Heenan, having preliminary plans prepared for a 1 story blimp manufacturing plant Aboretum Road. Estimated cost $40,000. Private plans.

N. J., Newark—Superior Air Products Co., 182 Malvern St., plans a 1 story addition to plant. Estimated cost $40,000. Architect not selected.

O., Cleveland—Ohio Crankshaft Co., William C. Dunn, Pres., 6,600 Clement Ave., awarded contract for a 1 story, 35 x 147 ft. addition to factory to J. L. Hunting Co., Guarantee Title Bldg. Estimated cost $40,000.

Tex., Beaumont—Commercial Pulp & Paper Co., c/o G. A. Kliman, have arranged to take over plant of Yellow Pine Paper Co. at Beaumont and plans improvements including new incinerators, disc evaporators, waxing machinery, etc., 10,500 tons of paper annual capacity. Estimated cost $125,000. Private plans.

Tex., Willow Springs—Arkansas Fuel Oil Co., c/o K. E. Jones, Shreveport, La., has work under way on construction of a gasoline refinery, 30,000 gal. capacity, six miles west of Longview. Private plans. Work being done by day labor. Some new equipment will be purchased.

Wis., Repon—Repon Handle Co. plans construction of a factory building. Estimated cost $45,000. Architect not selected.